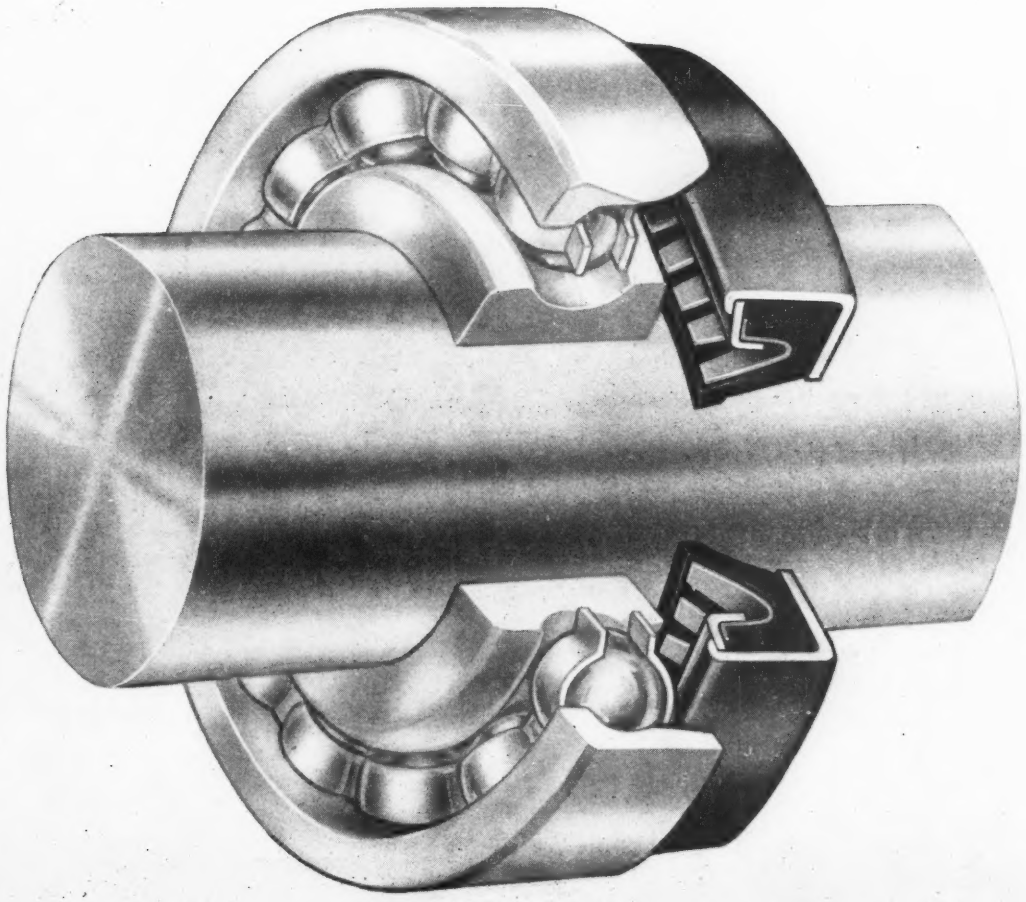
Practical Feature: SEALS and CLOSURES

AUGUST, 1953

Vol. 9, No. 4

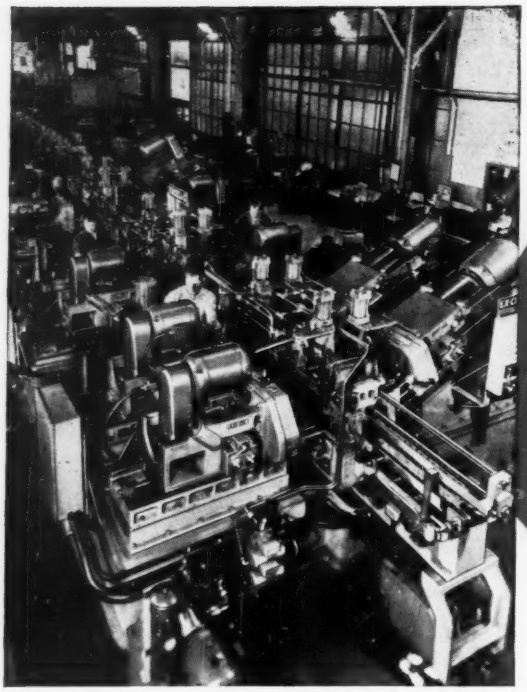
Lubrication ENGINEERING



OIL SEAL

JOURNAL OF

THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS



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Trabon

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used on transfer machines
in large automotive plant

this has a BEARING on the subject

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Oil and Grease Systems

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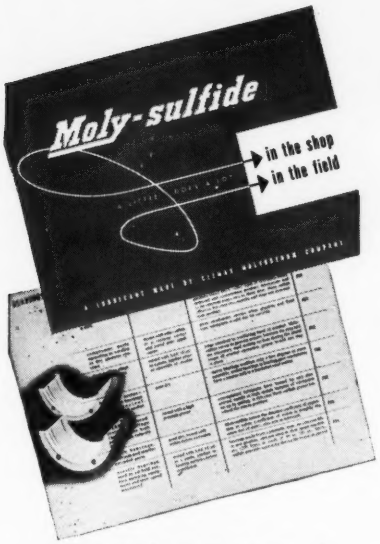

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Lubrication ENGINEERING

AUGUST, 1953, Vol. 9, No. 4

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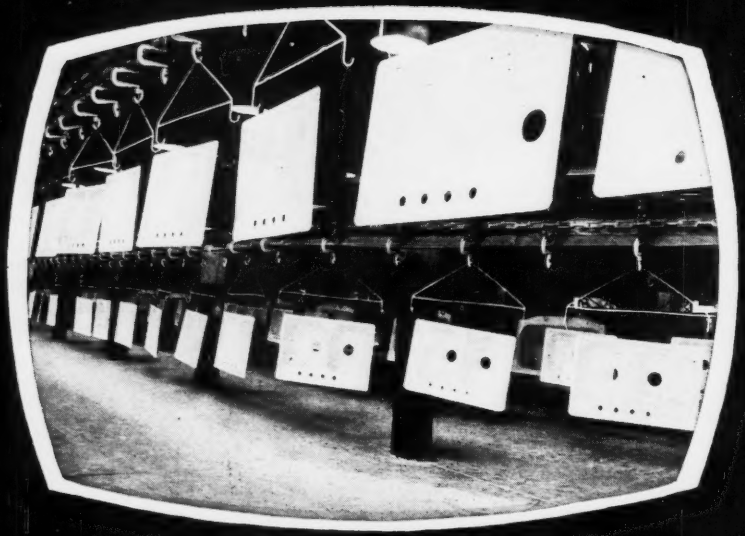
telling the story of 'dag' dispersions



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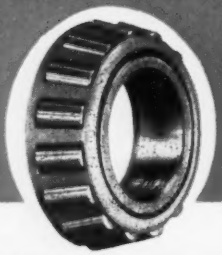

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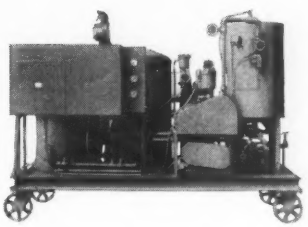

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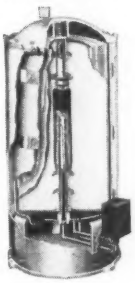

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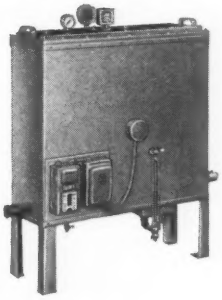

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advantages of Graphalloy Bushings . . .

LOW RATE OF WEAR

GRAPHALLOY bushings, both dry and submerged, have shown less wear than any other bushing materials by a factor of 8 to 1. They do not lose strength or wear away rapidly at elevated temperatures.

COMPRESSION CHARACTERISTICS

GRAPHALLOY is resilient and can be subjected to heavy interference fits so that shrink fitting, soldering, pinning, and clamping are unnecessary. Because of this characteristic, GRAPHALLOY can be pressed into a housing with enough interference fit to insure that the inside diameter of the bushing will expand at the same rate as the housing. Thus, if the housing and the shaft material have the same coefficient of thermal expansion, a constant clearance will be maintained between the ID of the bushing and the shaft. This characteristic is especially important for high temperature bushings, as in ovens and in seal rings on jet engines. It is interesting to note that, because it has a coefficient of thermal expansion about one-half that of steel, GRAPHALLOY is admirably suited for low temperature applications. The shaft and the bushing cannot seize since the shaft will contract more rapidly than will the GRAPHALLOY bushing.

GRAPHALLOY will stand considerable shock. Small bushings for airborne equipment have been subjected to loads as high as 50g.

LOW TEMPERATURE APPLICATIONS

GRAPHALLOY is especially suited for use in low temperatures where oil lubrication will solidify and cause sticking or where a bronze bushing will seize because it will contract more rapidly than the steel shaft. A special application is in liquid oxygen, where oil lubrication would be explosively dangerous.

GRAPHALLOY bushings are particularly suited for applications on signal equipment such as potentiometers, guided missiles, and rotating devices which must be operated over a range of —60° to +180°F without sticking or binding.

HIGH TEMPERATURE APPLICATIONS

GRAPHALLOY bushings operate satisfactorily in temperatures up to 700°F in oxidizing atmospheres in kiln and oven bearing applications where oil cannot be used because it will carbonize, and also in those instances where the oil fumes will contaminate the product.

Because GRAPHALLOY does not decrease in strength at elevated temperatures, it is especially useful in high temperature equipment where vibration is present and where steel bearings will soften and peen.

DRY APPLICATIONS

GRAPHALLOY is used often for room temperature applications where the bushings may be inaccessible, or where oil-lubricated bushings collect lint or fibre particles. The abrasive action of these foreign particles, which is a primary cause of excess bushing and shaft wear, is avoided by GRAPHALLOY bushings which do not introduce such particles into the bearing area. GRAPHALLOY bushings on difficult applications have run unattended for three to four years, in sharp contrast to the effort involved in maintaining a routine oiling schedule.

GRAPHALLOY bushings have been used where the bushing rotates because oil-lubricated bushings with this type of design tend to sling oil away from the shaft.

GRAPHALLOY is used for pistons and piston rings where oil cannot be tolerated. This is particularly true in compressors for instrument service where oil would eventually gum the small capillary tubes.

GRAPHALLOY is used for friction discs where the coefficient of friction must not change with wide swings of humidity and where torques must be maintained within narrow limits.

GRAPHALLOY can be used in small mechanisms where oil-lubricated bushings would produce too much drag. This is especially true for shafts having 1/16 to 3/8 inch OD.

GRAPHALLOY has been used for bushings in high load applications where there is vibration. If grease were used it would tend to be forced out of the area between the bushing and the shaft, allowing metal-to-metal contact. In this type of application GRAPHALLOY has been especially successful.

GRAPHALLOY has been used for turbine safety bushings. In this application, if the oil supply is lost, the shaft will run on the GRAPHALLOY bearing until the turbine can be brought to a stop without damage to the turbine blades.

SUBMERGED APPLICATIONS

GRAPHALLOY deposits a thin metal-graphite film on the shaft which is insoluble and therefore does not wash away. The thickness of the film is in the order of .00002". Because of this film, GRAPHALLOY bushings are used in water pumps, gasoline pumps, fuel oil pumps, spray pumps, meters, mixers, step and foot bearings; and are submerged in plating solutions, steam atmospheres, hot or cold fresh water, and sea water.

HIGH SPEED APPLICATIONS

GRAPHALLOY has been used for bushings and seals in high speed applications lubricated with water, gasoline, steam, or air. Speeds as high as 100,000 RPM have been maintained on ultracentrifuges with two inch diameter shafts. In these applications the GRAPHALLOY bushing will not seize on the shaft if the wall of the bushing is touched momentarily. This is an effective illustration of the advantage of a layer-lattice solid employed as a lubricant.

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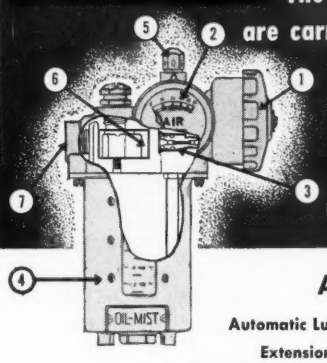
HOW IT WORKS: **The Oil-Mist Lubricator atomizes oil into microscopic particles which are carried in the air stream and distributed through tubing to bearings**



Oil-Mist airborne lubrication is accomplished this way: Compressed air entering the unit passes through air regulator (1) and air gauge (2). As this air passes through venturi (3) it draws oil from reservoir (4). Oil flow is set by knob (5). The mixture of air and oil from the venturi is thrust against baffle (6). Only the most minute, lighter-than-air particles are blown through outlet (7) into delivery line to lubricate bearings.

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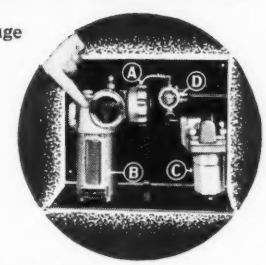

- Oil-Mist outlet ½" fem. p.t. Air gauge registers to 50 psi. Operating air pressure—5 to 20 psi.
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- Range of oils handled—to 1,000 sec. (S.U.V.) @ 100°F.
- Oil reservoir **(B)** capacity 12 oz. (approximately 1 week supply). Intake filter screen—70 mesh. Fill plug—9/16".
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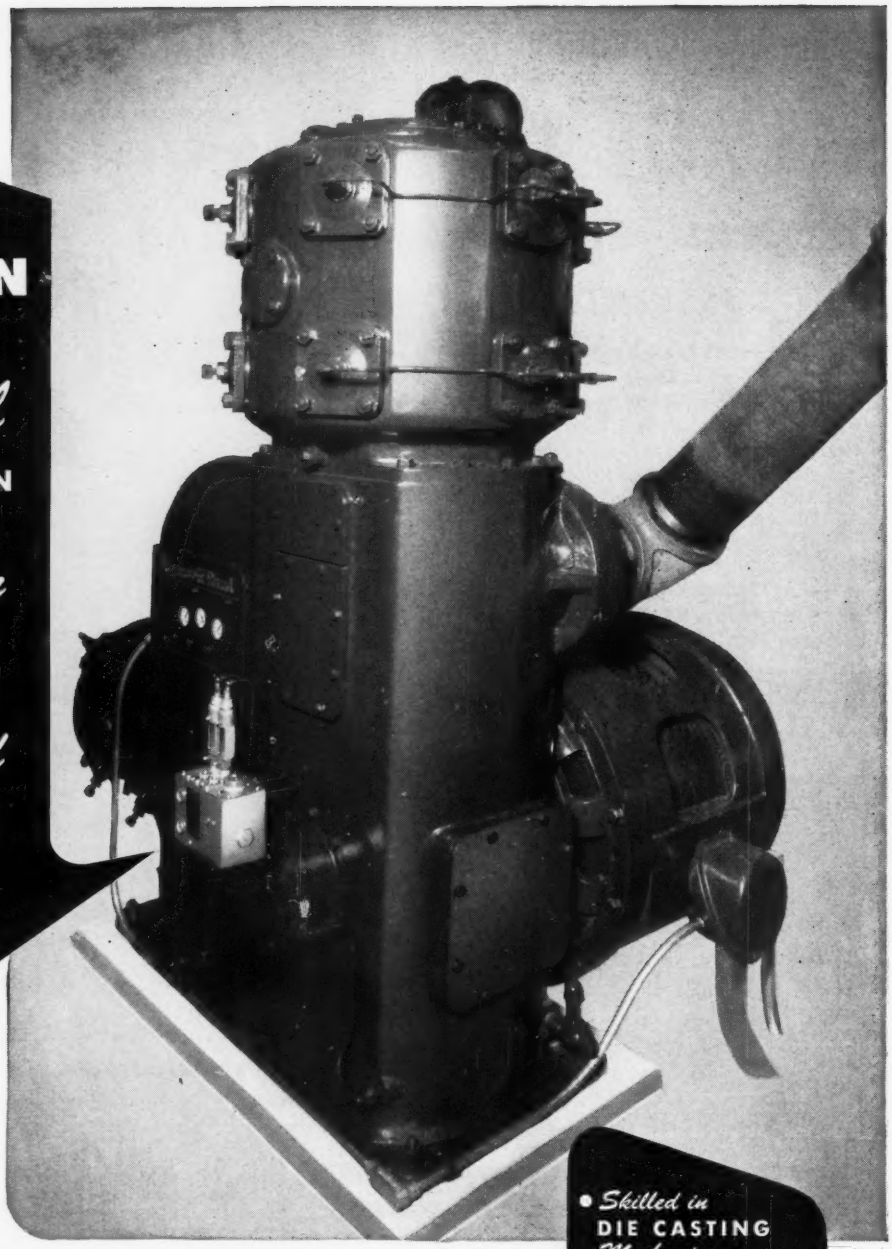

Illustrated, the Madison-Kipp Model SVH Lubricator on an Ingersoll-Rand XLE Compressor.

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- *Originators of Really High Speed* **AIR TOOLS**
- *Experienced in* **LUBRICATION** *Engineering*

....STANDARD EQUIPMENT ON COMPRESSORS, WORK ENGINES AND MACHINE TOOLS....

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HI-LITES OF FIRST ASLE TV PROGRAM: On May 17, 1953, the Cleveland Section presented the first ASLE TV program entitled "Life Depends on Lubricants" over NBC WNBK-TV in Cleveland, Ohio. Sponsored by The Cleveland Technical Societies Council and its forty-six member societies, these TV programs are presented as a public service under the general title of "Adventures in Engineering," and are financed by contributions from industry with the thought of attracting young people into an engineering profession. Participating in the program were: W. E. Campbell, ASLE President, Bell Telephone Labs.; C. C. Cantrell, National Refining Co.; J. Crankshaw, student, University School; R. H. Josephson, Cleveland Graphite Bronze Co.; W. Shafer, The Lubrizol Corp.; R. R. Slaymaker, Case Institute of Technology; and L. T. Unks, Sun Oil Co. Helping with the program were: A. O. Anderson, Aluminum Co. of America; J. L. Finkelmann, The Warren Refining & Chemical Co.; A. Nicol, The Ferro Enamel Co.; R. Sipple, Lincoln Engineering Co.; and N. I. Whitely, American Steel & Wire Co.

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A PERSONAL CHALLENGE: A job in the professional field is inherently carved to fit the individual. It can be as much or as little as one cares to make it. Satisfaction and rewards, moreover, are directly related to the interest and accomplishment. Lubrication Engineering is unusually responsive in this regard. It can and does range all the way from a haphazard squirting with a messy oil can to a position of highest responsibility in charge of preventative maintenance. The differences are highly personal and the position attained by a lubrication engineer is determined by his professional knowledge and his ability to sell himself. The accomplishments of leading men in this field attest this statement. While management may formally fix the status of lubrication engineering in a given plant, its position, in the last analysis, is largely that of an umpire. Management accords recognition and authority only as it is assured of a net benefit. Support is continued on the basis of results. The challenge is there. Take advantage of it. S.K.T.

NORTHWESTERN LUBRICATION ENGINEERING SHORT COURSE HI-LITES: Forty-two students attended the ASLE sponsored Lubrication Engineering Short Course at the Technological Institute of Northwestern University in Evanston, Illinois, June 15-24, 1953. Under the direction of Profs. M. F. Spotts & B. H. Jennings, Lecturers for the course included: C. A. Bailey, National Tube Co.; J. Boyd, Westinghouse Elec. Corp.; W. E. Campbell, Bell Telephone Labs.; C. R. Chapman, Standard Oil Co. of Ind.; D. M. Cleaveland, Bendix Aviation Corp.; S. D. Craine, W. A. Jones Fdry. & Mach. Co.; D. D. Fuller, Columbia Univ.; C. A. Geisinger, B. F. Goodrich Co.; W. P. Green, Armour Research Foundation; G. G. Lamb, Northwestern Tech. Inst.; O. D. Lascoe, Purdue Univ.; M. E. Merchant, Cincinnati Mill. Mach. Co.; W. W. Millett, Union Carbide & Carbon Corp.; A. E. Roach, General Motors Corp.; R. J. Schager, Cleve. Graphite Bronze Co.; J. B. Stucker, Pure Oil Co.; A. F. Underwood, General Motors Corp.; S. M. Weckstein, Timken Roller Brng. Co.; and T. R. Witt, Tennessee Eastman Corp.

Pictured above, from left to right, are (1st row): J. Tierny, C. A. Bailey, W. E. Elliott, M. F. Spotts, B. H. Jennings, W. P. Green, W. W. Merryman, E. M. Downs, M. Benett. (2nd row): R. O. Anderson, A. S. Koch, O. J. Riss, G. T. Lott, J. B. Tiernan, G. A. Darding, M. B. Peterson, E. H. Robinson, C. R. Chapman, F. G. Nierhoff. (3rd row): K. A. Woods, E. W. Brennan, R. H. Moth, M. M. Moore, P. W. Thompson, P. H. Goeltz, J. J. Stevens, G. W. Crosby, D. C. Spaulding. (4th row): W. H. Millett, W. D. Schroeder, M. A. Bigelow, E. B. Brenan, J. B. MacPherson, W. C. Smith, R. L. Peters, E. B. Rawlings, S. Abramovitz. Also attending were: R. Arnett, J. N. Bowden, F. T. Finnigan, J. W. Gilbert, W. D. Gilson, R. E. Kennemer, R. R. Laughlin, A. Manteuffel, R. Moyer, L. J. O'Brien, P. Pfeifer, and A. Zuhn.

9th NATIONAL CONFERENCE ON INDUSTRIAL HYDRAULICS: The Illinois Institute of Technology will be host to the 9th National Conference on Industrial Hydraulics, October 8 & 9, 1953, at the Hotel Sheraton in Chicago. Conference Director will be O. E. Teichmann, Armour Research Founda-

(Continued on page 217)

SEALS & CLOSURES *

by E. W. Fisher
The Garlock Packing Co.
Palmyra, N. Y.

Seals and closures are used to retain lubricants in bearings and to exclude foreign material from a variety of devices and mechanisms. There are two general classifications used for this purpose: (a) non-contact and (b) contact or rubbing type seals. The first group includes: (1) Straight, mechanical-clearances or plain type seals. (2) Annular groove type seals. (3) Labyrinth type seals. The second group includes: (1) Felt and cork type seals. (2) Non-spring loaded unit type seals. (3) Spring-loaded radially-acting type seals. (4) Spring-loaded axially-acting, mechanical or face type seals.

The straight mechanical clearance seal as shown in Fig. 1 is in reality not a true seal for it depends on the lubricant filling the space between the rotating and stationary members to effect a seal. The consistency of the lubricant and its ability to remain in this clearance are important factors controlling its effectiveness. These factors require grease lubrication and firm types of greases are preferred. Theoretically this type of seal might seem to be good for, since there are no rubbing parts, there is no frictional drag, no heat generated and thus no speed limitations. The fact, however, that a firm grease is required to make the seal effective, eliminates it from high speed service since most bearings require light lubricants for such speeds. If the bearing is pressure lubricated, this seal will act very favorably as a vent. Unfortunately, however, the breathing action of the bearing tends to force the lubricant out (leakage) and also to draw dirt in. Any dirt which is trapped in the lubricant distributes itself throughout the bearing housing and is most detrimental. It may be said, therefore, that the straight, mechanical-clearance seal is not particularly effective and its use is restricted to conditions where firm greases can be used and to relatively dirt-free locations where some lubricant leakage may be tolerated. Its lubrication requirements limit it to low speed or oscillatory service.

An improvement over a straight mechanical clearance is the annular groove or so-called "grease-groove" seal shown in Fig. 2. Interrupting the clearance with one or more grooves produces a turbulence in the seal slowing down its rate of leakage of lubricant. A drain-back, sometimes provided in multiple groove seals, makes possible the consideration of a fluid lubricant. However, both liquid and grease lubricants show some leakage. While annular groove type seals represent an improvement over straight, mechanical-clearance seals, they are still subject to the same general problems of leakage and contamination.

To make further use of dynamic forces which can be used for sealing, another step may be taken by complicating the clearance and creating a labyrinth as shown in Fig. 3. The labyrinth provides a longer, more torturous path of flow. With each change in direction of flow from axial to radial, centrifugal force becomes effective and tends to pump out any lubricant which enters the clearance. This in general is no disadvantage with semi-solid lubricants as only the excess will be pumped out, but it does mean leakage during this period. On the other hand, this same action tends to improve the seal efficiency against entry of extraneous materials.

A "slinger" may be created by placing one component of the labyrinth outside the housing with a consequent improvement in sealing against the entrance of extraneous fluids. The slinger may often be combined with other sealing devices with considerable success, but in all cases the speeds must be high to make these designs effective. Unfortunately, labyrinth seals are costly because they require a relatively large space, considerable machining is involved, and the assembly is complicated. They cannot be used under submerged conditions and the lack of definite sealing contact may require pressure balancing between seals in opposite ends of a bearing housing in such applications as in a fan or blower.

In an effort to provide a more positive seal and still retain compactness, simplicity and economy, the annular groove seal is sometimes modified by filling the groove with felt or cork. By thus eliminating the clearance, a totally different type of seal is created, called a "contact seal," and falls in the second general classification mentioned earlier. The name "contact seal" implies the use of two materials moving one against the other, a condition which would wear or burn away either or both materials if lubrication were not supplied.

*Presented at the ASLE 7th Annual Meeting, Cleveland, April 9, 1952.

In practice it is customary to prelubricate the sealing material and to rely on replenishment of lubricant from the bearing. The natural resiliency of the felt or cork must be relied upon to maintain sufficient radial loading to follow shaft gyrations and to maintain a sufficiently thin film of lubricant to prevent excessive leakage. Unfortunately the resiliency of felt and cork is not too enduring, particularly in presence of heat or moisture and the life of these materials is generally short. Felt has a further drawback of being porous. In practice, it is found that felt and cork seals are best adapted for use against grease and preferably limited to true running shafts where there is neither undue heat or moisture.

These seals are superior to clearance type seals in preventing the ingress of foreign materials but they do have certain weaknesses. Felt readily picks up foreign material which, if of an abrasive nature, may cause considerable shaft wear. Extraneous liquids may chemically attack the seals or cause shrinkage or rotting. The trapping of liquids may also promote corrosion and pitting of the shaft.

To further improve and protect these seals, additional grooves or a slinger or labyrinth may also be incorporated. Since the main purpose of such auxiliary devices is to exclude foreign material, they are placed on the atmospheric side of the contact seals. If the latter is felt, it must be provided with lubricant and so should be placed adjacent to the bearing to insure an adequate supply. Should a slinger be used with felt together with a fluid lubricant, there is a danger that the slinger may act as a pump and create excessive leakage. This condition can be relieved by installing another balancing slinger of the same size inside the housing. In this case the felt relies on oil fog for its lubrication.

Seals of the above type are complicated in assembly for a number of reasons. Solid rings of felt, in a solid housing, must be installed in place before the shaft is positioned and insertion of the latter may easily dislodge or distort the seal. When a split housing is used, there is always the danger of pinching the felt at the joint. Counterbored housings require end plates with screws or snap rings to retain them.

The next step to simplify the installation and to improve the sealing member is through the use of other materials. In regard to the latter, seepage through felt may be largely eliminated by the use of felt alternated with one or more layers of an oil-resistant rubber. The close fitting nature of the rubber ply and its natural resiliency contribute to a more efficient and longer lasting seal.

Installation may often be simplified by assembling the sealing material in a metal case which can in turn be easily press-fitted into a counterbored housing as is shown in Fig. 4. In a design of this type, exemplified by the unitary metal encased sealing member, it is now possible to use other materials such as leather or synthetic rubber. The close fibrous structure of leather reduces seepage and permits shaping a beveled lip which is more efficient than the broad soft contact of felt. Leather is quite resilient, hence against many heavy, viscous lubricants reasonable success may be obtained. However, leather, like felt, is attacked by many chemicals and will not withstand high temperatures, repeated wettings and, unless specially treated, it is subject to attack by fungi.

Synthetic rubber is an excellent alternate to leather and its use eliminates many of the variations and weaknesses found in the latter. The most common choice of synthetic is a Buna N type rubber. Neoprene may be used but, under some conditions, it has a tendency to corrode steel or to adhere fast to it during long periods of storage.

It is usual in thinking of rubbers to consider them elastic and to some extent this is correct. However, although they are more resilient than felts or leathers, they are subject to stress-decay and cannot be counted upon to maintain resilient, spring-like shaft contact for extended periods of time. The situation is aggravated by shaft whip or run-out and by elevated temperatures. Moreover, certain asphaltic base oils, solvents and synthetic lubricants may cause excessive swelling which causes loss of sealing contact. While this difficulty can be reduced by completely confining the rubber, except at the shaft, this arrangement incurs danger of shaft seizure.

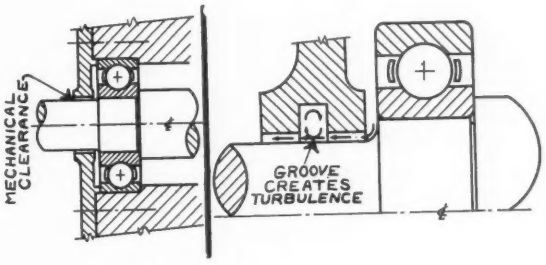


Fig. 1 (left), Fig. 2 (right).

Synthetic rubbers are superior for sealing against oil due to their many desirable characteristics. They are about the most resilient materials available and, where special conditions obtain, it is often possible to find or develop a special compound tailored to these conditions. They have excellent heat and aging properties and do not dry out in storage nor deteriorate in repeated wettings. In addition to these advantages, they are homogenous, non-porous, and can be molded.

The ability to mold synthetics opens a whole new field seal design through the introduction of a definite molded-lip type sealing-member whose cross-section may be precisely engineered to fit various applications. The first change is well shown in the simple assembly of a grooved sealing member in place of felt or leather as shown in Fig. 5. A design of this type allows fair stresses to be developed in the rubber and thus a tighter shaft fit is obtained. The groove provides the space into which the rubber can expand in event of high operating temperatures. If this space were not available and the rubber were fully confined, expansion from high temperature would cause seizure. The groove also provides the clearance for the lip to flex as it follows shaft movements.

Another important quality of synthetic rubber is ability to be bonded to metal. Simplified designs which take advantage of this property include those in which the sealing member is bonded into a metal shell, as shown in Fig. 6 (a) as well as those in which it is bonded to or around a metal insert

as shown in Fig. 6 (b). These seals have their chief value in being capable of production in very small cross section. The metal encased seal is made in sections as small as ⅛" square and is thus admirably suited for use with needle bearings. This small section permits the seal to be press-fitted directly into the same bore as the bearing without the need of counter boring or enlarging the outer diameter of the hub which are required by various other seal assemblies. Insertable rubber-covered metal seals, while not yet available with an equally small cross section (5/32" being about the minimum) have other important advantages. They will handle larger tolerances in the housing bore. The bore finish need not be as smooth. Thin walled housings or light alloy metal or die cast housings are not as likely to be split open. Further, the metal insert places the rubber under compression and insures a lasting, fluid-tight fit. Irregularities of the housing bore actually assist in locking the rubber in place and compensate for the gradual decay of stress in the rubber.

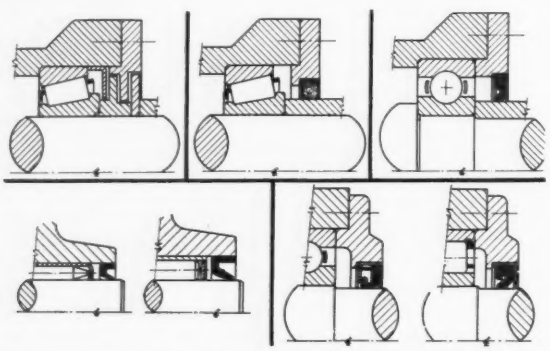


Fig. 3 (upper lift), Fig. 4 (upper center), Fig. 5 (upper right), Fig. 6(a) & 6(b) (lower left), Fig. 7(a) & 7(b) (lower right).

Because of advances in mold design, bonded seals also offer assurance of perfect concentricity. They also eliminate the possibility of leakage between the sealing member and the supporting metal member.

It is customary to consider rubber and synthetic rubber as resilient, elastic materials. However, it is observed that when they are under continual deformation the initial stress gradually disappears. As a consequence, an original snug fit of a sealing lip is gradually lost and if there is whip or shaft run-out, the lip may actually be deformed to larger than shaft size. Heat is an important element in hastening this type of failure and for this reason no-spring seals of synthetic rubber should only be used for light duty where temperatures and speeds are low and then preferably against viscous or semi-solid lubricants.

Stress decay and eccentric motion of shafts are two adverse factors which are always present in every seal application. To prevent leakage, a seal must follow every shaft gyration and an opening as small as a few ten-thousandths of an inch would cause troublesome leakage. Stress decay gradually robs the sealing member of its resiliency thus it fails to follow the shaft and leakage occurs. Shaft whip and shaft run-out can be minimized by careful bearing design with material improvement in seal life but cannot be eliminated entirely.

While it is not possible to eliminate stress decay, it can be minimized by placing the sealing lip under spring loading such as is shown in Figs. 7(a) & (b). Fig. 7(a) illustrates the first reasonably successful application of a spring to an oil seal. This seal consisted of an oak-tanned leather-flange with a coil or garter spring snapped around the outer portion of the axial lip flange. This combination was then clamped in a metal case whose cover retained the spring. However, the oak leather tended to become soft and porous in service and would lose its shape and permit the spring to "pop" out. Further, porosity of the leather permitted seepage. The next development was the oil-resistant composition molded seal shown in Fig. 7(b). This seal held its shape and was noteworthy for the introduction of the lug or finger-type serated spring. The spring was so assembled as to preclude its ever coming out and was positioned to place the load on the lip at the most advantageous point, directly over the tip of the lip. In contrast to a coiled spring, this spring was further unique in that it could not become plugged with dirt or corrosion products and thereby be rendered inoperative. The finger spring is always active since there is no possibility of the fingers becoming snubbed as occurs when the turns of a coil spring become even lightly embedded in the sealing member. On the other hand, the garter spring can be distorted far beyond service requirements without damage while a corresponding distortion of the finger spring could bend or set the fingers beyond an acceptable limit. While loading can be varied in either type of spring, extreme care must be used in making the coil spring to avoid undesirable variation.

As mentioned earlier, the first contact seals were made with hard oak leather. However, this material has poor heat resistance and so gave way to oil-filled leathers of double tannages. The sealing tip was made soft and flexible and, to prevent its collapse, springs were redesigned and loads lightened. Better leather plus lower operating temperatures due to lighter loads, stretched the life and expanded the field of usage. There are now available leathers impregnated with various synthetic rubbers or resins and these promise further progress in design, life and fields of usefulness.

The availability of synthetic, oil-resistant rubbers provided the ground work for vast improvements in composition seals. The first of these were the neoprene or chlorinated synthetic rubbers. However, these were found not to be too well suited for oil seals since they had a tendency to corrode or adhere to steel. Moreover, in high-speed work, they were prone to seize and burn. Fortunately, the Buna-N rubbers were developed soon afterwards. These were free from corrosive properties and were almost totally unaffected by most lubricants. The remaining weakness in synthetic rubbers is the lack of resistance to aromatic hydrocarbons and to many solvents especially those of the chlorinated type. With the availability of these new oil-resistant rubbers, the cross section of the composition seal was changed considerably to provide a thin flexible lip with a stepped or restricted area of contact, Fig. 8. The spring was also changed to take much more deflection.

Development of the synthetic seal also gave

rise to improvements in service. The smooth polished lip of this material was a most efficient seal. Unlike fibrous materials, its homogeneous structure did not permit seepage and, most important, it had better resistance to heat than other sealing materials. It was natural, therefore, to utilize the Buna-N rubber in seals with garter springs. Early designs were modified and attempts made to place the loading of a coiled spring at the optimum position over the tip of the lip of the seal. This design also incorporated the stepped lip and the thin heel or hinge section just back of the step.

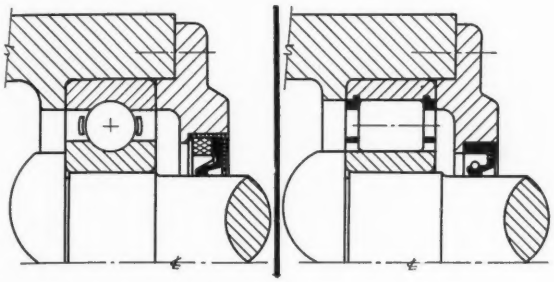


Fig. 8 (left), Fig. 9 (right).

There are many other variations possible in design. For example, in the case of the small seal, bonded to a metal case, a garter spring may be added provided a slight increase in size can be tolerated. Or seals with the rubber bonded around a metal insert may also be modified to include a spring, see Fig. 9. Progressing to large seals such as steel-mill roll-necks which range up to 48 inches in diameter, the seal shown in Fig. 10 can be used. It includes many elements of design. In sequence they are stepped lip, the thin heel or hinge section and finally the sealing member with a reinforcing U-shaped metal ring bonded into its flange. The basic spring is a stainless steel garter-spring, carried in a stainless steel finger spring designed to locate the load over the tip of the sealing member. The latter also provides a bed for coils so that snubbing is eliminated. The whole assembly is enclosed in a heavy steel case with a solid steel retaining or filler ring assuring a substantial and rugged unit.

Oil seals are sometimes inverted. A seal of this design is called an external seal and is usually built only for special installations where a conventional seal cannot readily be used. One important disadvantage is the care required in handling, since an exposed sealing member at the seal O.D. is easily damaged.

Self-aligning bearings present a special problem due to the type of shaft movement which they allow. For this service where the deflection is significantly large, a tangential seal is designed to make tangential contact to a spherically surfaced spacer, see Fig. 11. The center of this spherical surface must be at the bearing center to assure only sliding contact of the seal on its surface for all permissible gyrations of the shaft. This seal cannot be used for very high speeds.

In addition to these special designs, there are several more conventional combinations which deserve comment. For example, a felt ring may be inserted in the same case as a spring-loaded sealing member. The spring sealing-member holds oil while the felt excludes dust. A similar arrangement is achieved by using an unsprung sealing member in place of the felt. A further change may be made by putting a spring in both members. This seal is effective in keeping two liquids separate. In another arrangement an unsprung element, in front of a sprung member, serves as a baffle to keep coarse or heavy material from damaging the sprung member. If greater efficiency or some additional safeguard is considered necessary, for example, on a vertical shaft, it is possible to use two spring-loaded members both facing the same way.

In considering the use of combination seals, particularly those using two springs, it is necessary to remember that the speeds must be kept low if long service is to be realized. The reason is that these seals are acting as plain bearings running on a very thin film of oil and the generated heat will shorten seal life.

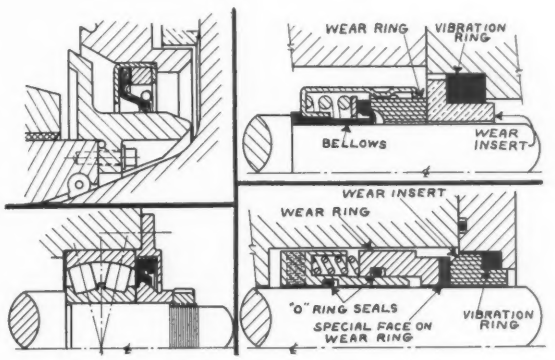


Fig. 10 (upper left), Fig. 11 (lower left), Fig. 12 (upper right), Fig. 13 (lower right).

Attention has repeatedly been directed to heat-resistance, an especially important subject in consequence of today's high speeds. The frictional heat is localized at the lip of the seal and at high speeds the temperature may exceed 200 or 300 F. Even a Buna-N rubber cannot stand 250 F. for long periods of time. It gradually oxidizes or post polymerization takes place. Both result in brittleness and finally small cracks appear in the wearing face of the seal and leakage then occurs. Although extremely flexible seals, which can respond to very light spring loads plus very narrow contact faces have minimized heat generation, there is need for synthetics with better heat resistance to make still better seals in the future. These are needed to extend present speed levels and to operate in locations where the ambient temperatures are high.

There are two new synthetics materials which have recently become available which offer exceptional heat-resistance. These are "Teflon" and "Silicone." The former was the first to receive consideration, but except for rare applications, it is too stiff to function well as an oil seal. Furthermore it is extremely expensive.

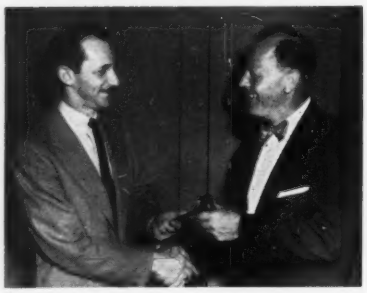
Silicone rubber has proved to be very useful since it will stand temperatures as high as 450 F. It is resilient and flexible and makes a satisfactory seal. However, Silicone seals do have some short-

comings. First, they are extremely weak physically, being easily nicked or torn and must be handled carefully. Second, they require new and lengthy curing methods which, plus the cost of the raw stock, further increases the cost of these seals. Third, their inherent chemical nature makes them susceptible to attack by silicone lubricants and light aromatic oils and such combinations must, therefore, never be used together. Continued development and research with silicone rubbers is constantly improving them and it is hoped that their most serious objections will eventually be eliminated.

There is one more type of seal which requires consideration. This is the so-called mechanical or rotary seal, see Fig. 12. Its principle or operation is again that of plain bearing running on a very thin film of lubricant. The great difference between it and the conventional oil seal is that the sealing faces are in a radial plane normal to the shaft axis. This difference no doubt assists in handling radial movement (run-out or whip) but axial shaft movement becomes a more critical element. Generally axial movement is compensated by a bellows or it may be by some more conventional packing such as "O", Chevron or "V" rings, as in Fig 13. Mechanical seals are most generally used as packing against pressures and it is often the magnitude of these pressures which determine the choice between the use of a bellows or packing rings. The ability of this type of seal to handle pressure is in no small part due to design which permits almost any choice of bearing materials, both soft or hard, for the contact faces since these my be solid, rigid members and may be readily lapped to any desired finish. The use of mechanical seals against oil in place of conventional seals is rather limited due to their higher initial cost as well as their space requirements.

In final summation it is well to recognize that where shafts turn, exact concentricity cannot be expected. This means run-out and where the substance to be sealed or excluded is liquid, the sealing lip must follow the shaft. In the ideal situation, the clearance between the shaft and seal will be no more than a thin film of oil. To best achieve this result, flexible sealing members with exterior spring-loading are the most effective. Shaft finish must be good, preferably on the order of 10 to 15 micro inches. Smooth, sharp-edged sealing lips are indicated for the seals. In high speed work, every consideration should be given to reduce the operating temperature to the lowest possible level. It is not possible to state a definite speed limit for seals for under ideal conditions there seems to be no limit. To overcome the ill effects of speed it is necessary to minimize heat generation, through the choice of lightly loaded seals and the provision for good lubrication, and to provide for effective heat removal or dissipation. There is a point, to be sure, where special consideration should be directed towards the choice of the seal and its installation. Sometimes this point may have been erroneously interpreted as the limiting speed for a seal. Bearing this in mind the following figures should be considered merely as a guide to the amount of attention that a given application should receive. Heavy duty, single, dual or combination spring-loaded seals should be satisfactory for low and medium speeds of about 1000 feet per minute. Non spring-loaded seals and most of the modern lightly loaded single seals should be satisfactory up to about 2500 feet per minute. If an application involves temperatures unavoidably or exceptionally high, silicone should be considered (provided there are no silicone or aromatic fluids present to get in contact with the seal). In critical sealing applications the mechanical seal should receive consideration provided there is ample space and the installation will bear the added cost. If pressures of sufficient magnitude are present, a mechanical seal may be a necessity.

Section News



BALTIMORE: May meeting — Annual installation of officers (See ASLE Directory; shown above is incoming Chairman W. C. Landis (right) receiving the gavel from retiring Chairman W. Rosenberg), together with an address by A. Chapman, Arabian-American Oil Co., on International Oil & World Politics. (Submitted by W. W. Witmer, Sec'y-Treasurer.)

CHICAGO: April meeting — Annual election of officers (See ASLE Directory), together with an address by J. D. Krummell, A. T. Kearney & Co., entitled How Planned Lubrication Can Lower Plant Maintenance Cost.

May meeting — J. S. Aarons, U. S. Steel Corp., presented a paper entitled Laboratory Testing Of Lubricants, illustrated with colored slides.

June meeting — Golf tournament followed by a dinner at the Cog Hill Country Club, with G. Findlay, J. W. Hopkinson, R. H. Josephson, and W. C. Kesler as guests. (Submitted by A. B. Two, Vice-Chairman.)

CLEVELAND: (See Lube Lines: Hi-Lites of First ASLE TV Program. Submitted by C. C. Cantrell.)

CONNECTICUT: May meeting — Annual election of officers (See ASLE Directory), together with an address by M. C. Shaw, Massachusetts Institute of Technology, entitled Grinding. (Submitted by A. H. Hoge, Vice-Chairman.)

DAYTON: June meeting — Annual election of officers (See ASLE Directory. Submitted by E. C. Briggs, Sec'y.)

FORT WAYNE: June meeting — Annual election of officers (See ASLE Directory. Submitted by R. L. Gruenert, Retiring Chairman.)

LOS ANGELES: April meeting —

(Continued on page 223)

SOLID LUBRICANTS*

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INTRODUCTION: The rapid increase in the use of complex mechanisms for military applications has led to an increase in interest in solid lubricants. These mechanisms are frequently required to operate over an extremely wide temperature range that from -100 F. to 750 F. is being discussed. In many cases mechanisms having inaccessible bearings may be stored for indefinitely long periods, and yet must function with split-second accuracy at a moment's notice. Under these circumstances dust collection and corrosion become important considerations in addition to that of lubricant stability. It is difficult to see how all these requirements can be met by lubricants based on organic fluids. Progress in research in recent years gives promise that these problems can be solved by the use of solid lubricants.

Since there seems to be no comprehensive classification and discussion of solid lubricants available in the recent literature, it is the purpose of the present paper to provide such a review.

A solid lubricant may be defined as a solid material interposed between two relatively moving surfaces to prevent contact between them and thus to reduce friction and wear. To do this with maximum effectiveness all sliding must be between layers of solid lubricant. Thus to understand the ideal properties to be sought in a solid lubricant, a knowledge of the modern theory of solid friction should be helpful.

THE MECHANISM OF SOLID FRICTION: According to the widely accepted theory, as developed by Holm[1], Merchant[2], and Bowden[3], when two plane surfaces are brought in contact with a moderate pressure of, say, 100 lb. per sq. in., the real area of contact is only a very small fraction of that calculated from the macro dimensions, that is, the dimensions of the surface of contact, taken as a plane. Pressure welding of the two surfaces takes place because of the high pressure on the small real area of contact and when relative motion takes place the friction force is predominantly that necessary to shear these welded junctions.

This is because the smoothest surfaces, on a molecule's eye view, look like the Rocky Mountains. This is clearly seen from Fig. 1, which is an electron micrograph, 9900 magnification, of a typical portion of a replica of a steel surface which has been polished to a metallographic finish — a finish so smooth that no scratches are visible at 100X magnification through an optical microscope. When it is realized that a molecule of iron oxide, which is the material of which the surface of a steel bearing is composed, has a diameter of only one one hundred millionth of an inch (2.5Å), and that the faint scratches in the lower right hand corner of Fig. 1 have a width approximately two hundred times as large as this (500Å) it will be clear that to the molecule the surface is very rough indeed.

Thus, when the two plane surfaces are brought together, they touch, on initial contact, at relatively few points, where peaks of asperities coincide; if they are surfaces of the commonly used base metals, steel and bronze, a highly magnified portion of their cross section will appear somewhat as illustrated in Fig. 2, the oxide layer shown having a thickness of the order of four to eight one hundred millionths of an inch (10Å to 20Å). The pressure on peaks in contact is enormous, so that the yield pressure of the metal is exceeded and it flows plastically. Since the oxide film is brittle, it cracks at weak spots, and metal is extruded through the cracks; where extruded metal comes in contact on the opposing surfaces, pressure welding takes place. This process continues until sufficient area of contact has been created to carry the load without further metal flow, i.e. until (1): $W = Ap$, where W is the normal load, A is the real area of contact and p is the flow pressure of the metal. Bowden and Tabor show by means of contact resistance measurements, that the real area of contact is independent of the apparent area, as determined from their macro dimensions, of the surfaces in contact, and over a wide range, of their roughness, being only dependent on the load according to equation (1) above.[3] An idealized picture of what might be the appearance of a cross section of the contact between two opposing peaks such as those circled in Fig. 2, at equilibrium, is shown in Fig. 3.

Now, if relative motion is imparted to the surfaces, the welded junctions are sheared, new ones created and sheared, and mutual ploughing action takes place. The force required to do the weld-shearing and the ploughing, plus the additional force to lift interlocking asperities over each other, is the friction force. Bowden[3] has shown that for most practical purposes the predominant force is that necessary to shear welds; although his picture still lacks perfection it explains most of the phenomena of friction very satisfactorily and has been supported in recent years by an impressive volume of experimental evidence. For film-free metal surfaces, his theory leads to the expression (2): $F = As$, where F is the friction force, A the real area of contact, and s the shear strength of the welded junctions. If now equation (2) is divided by equation (1) it is found that (3): $F/W = f = s/p$, where f is the coefficient of friction. That is, the friction force is directly proportional to the load, which is known as Amontons' Law.

It is clear from the picture presented in Fig. 2 that an appreciable fraction of the real area of contact is oxide film. If this fraction is a, then (4): $F = A\,[as_o + (1-a)\,s_m]$, and, from (1) and (4), (5): $f = as_o/p_o + (1-a)\,s_m/p_m$, where s_o and s_m are the shear strengths of the oxide and metal respectively.

When solid surfaces are in rubbing contact, the friction force is approximately independent of the speed, a relation which is consistent with the above theory.

THE PROPERTIES OF SOLID LUBRICANTS: It is now possible to deduce the desirable wear and friction reducing properties of a solid lubricant film: (1) It should have low shear strength. (2) It should adhere strongly to the surface. (3) It should have good elastic properties. (4) It should be continuous.

In addition to the properties of the solid lubricant given above, which directly influence its frictional behavior, there are several others which are of practical importance in choosing the type of film for a given function. These are:

1) THERMAL STABILITY: One of the most important uses of solid lubricants is for reducing wear at high temperatures; high thermal stability is a must in these applications.

2) MELTING POINT: This property is of importance for the reasons given under 1); in addition it is important because organic solid lubricants lose much of their friction reducing power above their melting point[3]. This is because their solidity is due to the strong lateral attraction between the molecular chains composing them which, combined with low shear strength, makes them effective solid lubricants. Above the melting point the lateral attraction is destroyed by thermal motion, and the metal surfaces are no longer prevented from contacting and welding at high points.

3) ELECTRICAL CONDUCTIVITY: High electrical conductivity is needed when solid lubricants are used to reduce wear of sliding contacts. When insulators are subject to rubbing contact, a solid lubricant of low conductivity must be used. The wide variety of solid lubricants available makes it possible to select a lubricant having high or low conductivity according

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to need.

4) THERMAL CONDUCTIVITY: This property is important in consideration of the use of plastics as solid lubricants or bearing materials. Because of their poor thermal conductivity, surface temperatures generated by friction cause local melting and balling-up of material on the surface under certain conditions. Mixing of metal powder with the plastic to improve the thermal conductivity promises to give long-life bearing material which will operate with low friction and no galling without lubrication by conventional fluid lubricants[8].

5) CHEMICAL INERTNESS: Solid lubricants are useful when reactive chemicals or their vapors have access to the bearing.

6) CORROSION PREVENTIVE ABILITY: Since solid lubricants are frequently called on where inaccessible parts are required to function intermittently, with long inactive periods in between, for indefinitely long times, corrosion of the parts must be prevented. If the film satisfies the friction-reducing requirements mentioned earlier as well as item 5) in this group, it is likely to be effective in preventing corrosion.

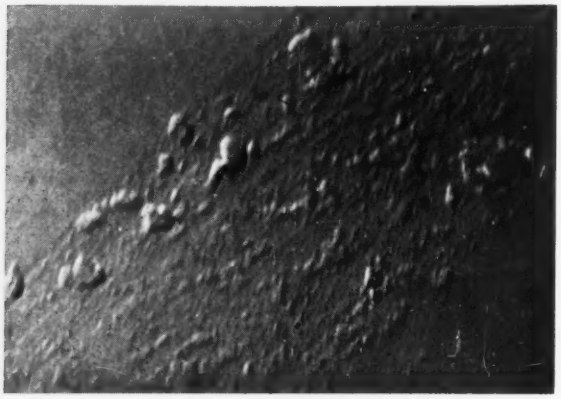

Fig. 1 Electron micrograph of silica replica of a typical portion of a metallographically polished steel surface showing asperities in the surface; magnification 9900X.

7) COLOR: The color is sometimes a consideration where it is extremely important to avoid soiling of processed material.

8) DENSITY: For solids which are applied in the form of dispersions in solvent, low density material makes it easier to maintain a stable dispersion.

9) STATE OF SUBDIVISION: For a uniform, continuous film applied for a dispersion in solvent, a low and uniform particle size makes for more stable dispersions and greater uniformity of the applied film.

10) FREEDOM FROM ABRASIVE CONTAMINANTS: This property speaks for itself — it is of especial importance when use is made of naturally occurring solids which may be contaminated with silica or other abrasive minerals.

There is one important characteristic which solid lubricants as a class lack — the ability to self-heal breaks in the film. They function successfully only as long as the film remains continuous. Since the relatively moving layers of solid lubricant are going through the same processes of weld-breaking, pickup, and transfer as occur between high friction surfaces, although at a much slower rate, the life of the film is not indefinite and its end is characterized by rapid increase in friction and serious galling of the underlying metal. Research designed to extend the life of solid lubricant films as far as possible is therefore important. It is also important to design equipment employing solid film lubricants well within the life span of the lubricant. Unfortunately there is no standardized test method available for evaluating the life of a solid lubricant, although preliminary discussions aimed at developing such an instrument have been held by the Coordinating Research Council, Inc. Because of the rapidly increasing use of solid lubricant films, a standardized method for their evaluation is much to be desired.

CLASSES OF SOLID LUBRICANTS: For convenience in classification, solid lubricants may be grouped into six classes as follows:

1. Layer-Lattice or Laminar Solids: Since the members of this class are almost all inorganic materials, most can withstand high temperatures, and several are unusually inert. They are therefore useful in a wide variety of applications, and are relatively well known. Examples are graphite, molybdenum disulfide or molybdenite, popularly known as moly-sulfide, tungsten disulfide, vermiculite, mica, boron nitride, borax, silver sulfate. The oldest and best known is graphite, but in recent years molybdenum disulfide has increased rapidly in popularity. All these materials have crystal lattices consisting of layers, the bonding forces between the atoms within the layers being much stronger than those between adjacent layers, which are relatively weak van der Waal's forces, so that the shear strength of the solid is low[9].

In the case of graphite, the ease of slip between adjacent crystal planes has been shown to result from the presence on the surface of an adsorbed water or oxygen film. In nitrogen at a relative humidity lower than 5%, or in high vacuum, rapid wear with very high friction takes place [10, 11, 12, 13]. If the graphite is run-in under normal conditions, so that sufficient water is adsorbed on the layers to give low friction, it becomes highly oriented with the slip-layers flat on the surface. Once this orientation is accomplished, wear is very slow, as for graphite in a normal atmosphere, down to a relative humidity as low as 0.4%.

The rapid wear of graphite in vacuum is readily reduced by incorporation in the powder mix or dispersion of a small amount of a variety of lubricants or adjuvants, as they are known. For most practical purposes almost any heat-resistant liquid or high-melting solid organic lubricant will do. For use in vacuum under conditions where the rubbing surfaces are hot enough to boil out or carbonize organic adjuvants, inorganic adjuvants such as barium fluoride[11] and lead iodide[12] have been developed to give desirable performance.

The slipperiness of molybdenum disulfide is not reduced in vacuum, but it cannot be substituted for graphite in brushes operating under high altitude conditions and heavy current loads, because the local surface temperatures are high enough to oxidize the MoS_2 to MoO_3, which is highly abrasive.

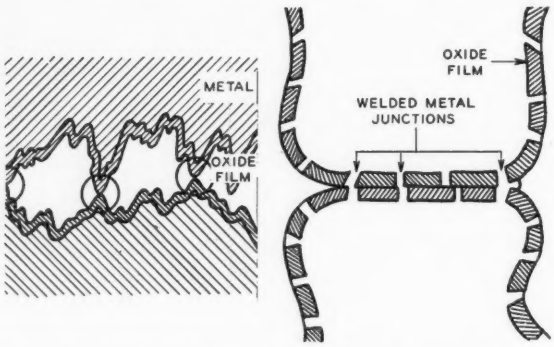

Fig. 2 (left) Idealized representation of a highly magnified cross section of a portion of two plane surfaces at time of initial contact.

Fig. 3 (right) Idealized representation of a highly magnified cross section of tips of two opposing peaks of two plane surfaces in contact at equilibrium.

A rubbed film of MoS_2 becomes highly oriented in the same manner as graphite[9].

Both graphite and molybdenum disulfide have high thermal stability. Graphite does not begin to oxidize at an appreciable rate below 350 C., when it is converted to carbon dioxide, a harmless gas; in vacuum or an inert atmosphere it is stable to temperatures above 1000 C. Molybdenum disulfide is stable in air to 350 C., and in vacuum or in an inert atmosphere to at least 600 C.[28] The decomposition temperatures in air are greatly dependent on the state of subdivision of the solid lubricant and on the nature and amount of bonding agent present.

They are both chemically inert, graphite having the edge in this respect, but give little or no protection against corrosion when rubbed into the surface from the powder.

Rubbed films are also relatively ineffective as wear reducers, presumably because they are not thick enough and do not adhere well enough to the surface; but they are very effective at reducing friction. They are useful for lubrication of parts undergoing intermittent operation at infrequent intervals, where friction reduction is of paramount importance. Molybdenum disulfide appears to have the edge for this type of application; it is said to adhere more strongly to metal surfaces than does graphite.

Graphite is obtainable in a much lower and more uniform particle-size range than other layer-lattice lubricants; because of its much lower density its dispersions have greater stability towards separation than molybdenum disulfide dispersions.

Tremendous increases in the wear life of both graphite and

molybdenum disulfide can be produced by bonding them with a suitable resin which provides the needed adhesion to the surface, the resin-bonded solid particles being dispersed in a suitable solvent to give a liquid finish which can be painted or sprayed on to the surface requiring protection. Effective formulations of this type have the additional advantage that they provide excellent protection against corrosion. Godfrey and collaborators have successfully used carbonized corn syrup, silicone and asphalt-base varnishes, and glycerol resins as bonding agents and have shown that the mechanism of bonding is the same for a variety of layer-lattice solids[14, 15, 16]. Other bonding agents which have been used successfully are alkyd resins, phenolic resins, epon resins, and methyl methacrylate resins.

With the aid of suitable driers, successful air-drying, bonded, solid films can be made, but for long life at heavy duty a baked film should be used.

Except for the work of Godfrey and collaborators[14, 15, 16], very little quantitative work has been done on the wear properties of these bonded, solid lubricants. Some preliminary work in the Bell Laboratories indicates that particle size and distribution, purity of solid lubricant, and ratio of quantity of solid lubricant to binder are all very important in determining the life of the film. This work was done on a Taber abraser using a 3/32" thick, 2½" diameter steel disc, having a rounded edge, in place of each of the two calibrase wheels. These discs were firmly clamped in position so that only sliding motion against the specimen plate was allowed. The solid lubricant mix was painted or sprayed on a steel or brass specimen plate which was clamped in position. A 500 g. load was used up to 10,000 revolutions; beyond this point a 1000 g. load was used. The film was reported as having failed when base metal was visible in the wear track.

In Table 1 and in Fig. 5 are given the results of a determination of the effect on film life of graphite-resin ratio with two different synthetic graphites using a methyl methacrylate binder. The dispersions were painted on a polished brass plate from a 30% dispersion in cellosolve acetate to give a film approximately 0.002" thick. The film was baked at 300 F. for 1 hour.

	No. of Revolutions to Failure	
% Graphite by volume	Graphite Type 1	Graphite Type 2
10	20,000	.....
15	.....	30,000
20	63,000	.....
22.5	.....	76,500
30	950	28,000
50	200	22,000

Table 1—Effect of increase in graphite-resin ratio on wear resistance of resin-bonded graphite film.

It will be observed that for each mix there is a critical composition which gives maximum wear resistance. Below this maximum the relatively poor wear characteristics of the resin become evident; above it there is insufficient binder and the film powders away rapidly. Although the method used was relatively crude, it is believed that the general effects observed are significant. Good results were also obtained in these studies with alkyd resin binders, but for reasons which are not clear, mixes containing molybdenum disulfide gave unsatisfactory results with either type of resin.

Pretreatment of the metal surface by methods which improve the adherence of protective finishes increases the life and corrosion-protective ability of bonded solid lubricant films. In Table 2 and Fig. 6 are given data on the life of films on treated and untreated metal surfaces carried out on a Taber abraser as outlined above, but using sprayed films 0.0005" thick which were much more uniform and accurately controlled than those tested in the previous runs.

It is believed that there are interesting possibilities in these bonded solid lubricant films, and carefully controlled basic work is badly needed on such factors as base metal treatment, type of solid lubricant and bonding agent, ratio of solid lubricant to bonding agent, particle size of solid lubricant, heat treatment of bonding agent, etc., so that suitable basic formulations for various service needs can be established.

In addition to their use in films on surfaces, these materials are used as mixing agents to improve the wear resistance of other solid materials. For example, the addition of graphite improves the wear resistance of hard rubber, laminated phenolics, asbestos, and metals and alloys.

Perhaps the most striking example of the use of a layer-lattice solid lubricant in combination with other materials is the admixture of metals with graphite to make sliding electric contracts and self-lubricating bearings of various types. Two different procedures are in general use. The first, used mainly for conferring improved strength and electrical characteristics on carbon brushes, is as follows: graphitized carbon and metal powder are mixed in the desired proportions, the mix subjected to high pressure in a mold, and heated to the sintering temperature. In the other process the compressed, graphitized carbon is impregnated with metal to make a wide range of sizes, shapes, and strengths of self-lubricating bearing materials which are used at high temperatures, in reactive atmospheres, and in inaccessible locations. Silver-impregnated graphite made in this way is an effective self-lubricating contact material for sliding contacts making and breaking high currents. There is no reason to suppose that molybdenum disulfide and other effective layer-lattice type solids cannot be employed in this and other ways to improve the wear resistance of various solid materials.

Layer lattice lubricants are also effective additives to liquid lubricants for improving their lubricity. It is the author's opinion that in this use relatively little benefit is obtained from the addition of small percentages of solid. For maximum effectiveness the percentage of solid should be at least 10.

Solid Lubricant	Metal	Metal Treatment	Revolutions to Failure
Graphite-Acryloid	Steel	Polished	950
Graphite-Acryloid	Steel	Phosphated	85,000
Graphite-Acryloid	Brass	Polished	18,500
Graphite-Acryloid	Brass	Oxalate Film	65,000

Table 2—Improvement in wear resistance of bonded graphite—film by chemical treatment of metal surface.

2. Solid Organic Compounds—Soaps, Waxes, and Fats: Examples of this class are metallic soaps, the most common of which are aluminum, calcium, zinc, magnesium, sodium and lithium stearates; waxes, such as microcrystalline waxes, beeswax, spermaceti wax; synthetic waxes such as high molecular weight alcohols, amides, esters, etc.; solid fatty acids, such as stearic and palmitic acids; and fatty esters such as lard and tallow.

Some materials in this class possess self-healing ability, although the rate of repair of the film is much slower than with liquid lubricants and greases. Probably, for this reason, they are effective lubricants for light-duty operations below their melting points even when their adherence to the surface is relatively poor as in the case of non-polar type materials such as micro-crystalline waxes. The materials which are more chemically active give excellent wear and friction reduction, even above their melting points on base metals, by reacting with the surface. Thus stearic acid gives very low friction and good wear reduction on a variety of metals up to temperatures corresponding to the melting point of the metallic stearate. The soaps are all excellent friction reducers but cannot be as readily applied to the surface as the fats and waxes because of their higher melting points and relative insolubility in common solvents.

Most of the waxes and fats can be applied from a solvent. They are also applicable from a melt, or may be rubbed over the surface.

They can be incorporated in a solid to improve its lubricity. Mixing of paraffin in soft rubber reduces its friction and wear against soft rubber or other plastics, whereas graphite and molybdenum disulfide are relatively ineffective when used in this manner. The paraffin is insoluble in the rubber and has sufficient mobility to bleed out slowly to the surface, renewing the lubricant film there as it is used up. The soaps are widely used in the powder mix in molding and forming operations with resins, ceramics, metal powders and in tablet manufacture. The soap acts as a lubricant to facilitate slip between particles as well as a mold release lubricant. Paraffin in trace quantities is excellent for lowering friction and wear between textile fibres[29].

The lowering in friction between phenolic resin and phenolic resin produced by variation in the amount and nature of the soap-type molding lubricant in the mix, may have an important and interesting bearing on the shatter resistance of a finished telephone handset. In assembly of telephone handsets the ear and mouthpieces are threaded onto the body with a torque wrench set to a figure high enough that unthreading of these parts will be difficult. If the friction between the parts molded from a given mix is appreciably lower than the norm, the threaded parts will be screwed up tighter with the standard torque and the body thus subjected to greater stress. Recent tests have shown that variation in molding lubricant content of the handset mix can produce very large differences in the stress produced in assembling the finished handset.

Perhaps the most extensive use of this class of lubricant is in the wire drawing and metal forming industry. All classes of solid lubricants find use in these operations, but organic solids, no doubt because of their price advantage, are very commonly used. The compounds are generally used with a carrier such as lime or a phosphate film on the metal, and it is interest-

ing to note that, as predicted by the theory given above, the melting point of the mix is found in practice to be an important characteristic, higher melting point mixes being employed for coarser wire, heavier drafts, and higher speed drawing[17].

Wax deposited from a 20-30% solution makes a satisfactory running-in lubricant for gears and other lightly loaded parts of instrument mechanisms, which generally require no further lubrication; microcrystalline wax blended with spermaceti wax or with small percentages of stearic acid can be used.

Where low friction is desired for only one operation after an indefinite storage period, a light wax film has desirable properties. A good example of this type of operation is in the lubrication of the wire in the cable lashing machine pictured in Fig. 7. This machine pays out wire which lashes heavy cable to a supporting stranded steel wire. In trials of this machine the wire developed snarls as it unreeled. A light coat of a high melting point synthetic wax on the wire eliminated this difficulty.

3. Chemical Conversion Coatings: The best known films in this class are sulfide, chloride, phosphide, phosphate, oxide, and oxalate films. The first four types are generally formed in situ at points of incipient seizure as a result of reaction between additives in an extreme pressure lubricant and the surface of heavily loaded rubbing parts. Fig. 8 shows the effect of increasing the thickness of a sulfide film on copper on the coefficient of static friction between two copper surfaces. It will be observed that the coefficient of friction reaches a minimum at a thickness of about 1000Å under the experimental conditions employed[18]. Contact resistance measurements indicate that on the horizontal part of the curve all metal to metal contact has been eliminated[19]. Table 3 shows the friction reduction produced by oxides and sulfides on steel, copper and brass.

Metal Combination	Treatment of Surfaces	Static coefficient of Friction, μ Untreated Surfaces	Treated Surfaces
Steel-steel	Oxide	0.78	0.27
Steel-steel	Sulphide	0.78	0.39
Brass-brass	Sulphide	0.88	0.57
Copper-copper	Sulphide	1.21	0.74
Copper-copper	Oxide	1.21	0.76
Steel-steel	Oleic acid	0.78	0.11
Steel-graphite	Oleic acid	0.21	0.09
Steel-steel	Acto oil	0.78	0.32
Steel-steel	Oxide, Acto oil	0.78	0.19
Steel-steel	Sulphide, Acto oil	0.78	0.16

Table 3—Friction-reducing effect of solid films on steel, brass, and copper.

It will be observed that the friction for clean steel lubricated by an oil of poor boundary-lubrication power (acto oil) is lowered 40% by the presence of an oxide film and 60% by a sulfide film on the surfaces. The close parallelism of the measured friction lowering between chip and cutting tool when lubricated with liquids forming chlorides and sulfides at the rubbing surface, and that calculated from the shear strength of the solid film is demonstrated by Merchant[20].

The thin film of oxide which occurs on all base metals, even immediately after cleaning by abrasion or cutting, acts as a solid lubricant, although not the best, because of its poor elastic properties and relatively high shear strength. Nevertheless, were it not for its presence, seizure and galling would be very serious problems[3, 4, 5]. If the oxide film is thick enough, metallic contact can be completely prevented and friction appreciably reduced, as is shown for a sulfide film on copper in Fig. 8, see Table 3.

From the model in Figs. 2 and 3 it would be expected that, at very low loads, the distortion of the surface would not be sufficient to break the film. The work of Whitehead[8] shows that this is so. Fig. 4, taken from Whitehead's paper, shows that at very low loads the coefficient of friction is low and constant, following Amontons' Law of friction; at a critical load it becomes erratic and rises sharply to another constant value. In the region of low loads, the friction has been shown by contact resistance measurements to be caused by the shearing of oxide bonds[7]. In the highfriction region, the friction is now caused largely by the shearing of metal bonds, although the oxide film still carries an appreciable fraction of the load, since when measurements of friction are made on clean metals thoroughly outgassed in high vacuum, complete welding at all points takes place, and sliding is impossible[8]

Phosphate and oxalate coatings, are used as a supplement to other lubricants, both liquid and solid. The phosphate coatings, in particular, have extensive use in a variety of operations in forming and working of metals, usually in combination with other lubricants[21, 22]. They are also widely used to assist run-in, particularly in applications where scuffing is liable to occur[23].

Phosphate coatings consist mainly of zinc phosphate mixed with small amounts of iron and manganese phosphates in varying proportions depending on the application. They can be applied to ferritic iron alloys and to aluminum and zinc alloys. The chemistry of the process is discussed by Darsey and Cavanagh[24].

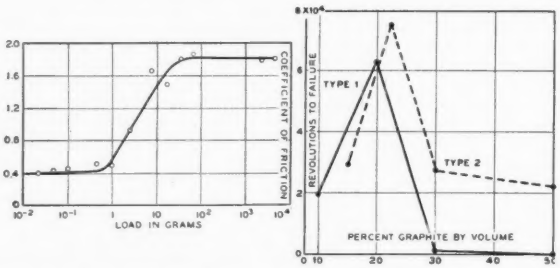

Fig. 4 (left) The effect of load on the coefficient of friction of copper on copper (after J. R. Whitehead[8]).

Fig. 5 (right) The change in wear resistance of a baked resin bonded graphite film on polished brass with change in graphite content for two types of synthetic graphite.

When chemical conversion coatings are used in combination with other solid lubricants, a surface is produced which outwears by a very large factor either lubricant alone. An example of their use with bonded graphite has already been given. Barwell[23] shows that striking reduction in scuffing wear and increase in seizure load can be produced by using bonded molybdenum disulfide on a phosphate coat, and paraffin is shown to be almost as effective when supplemented in the same manner. In addition to acting as a low-shear buffer, the porous phosphate coat absorbs lubricant and feeds it to the rubbing areas.

4. Soft Metals: The extent to which a thin film of a soft metal on a hard metal substrate reduces friction has been demonstrated by Bowden and Tabor[3] who have obtained coefficients of friction as low as 0.04 for an indium film 1.6×10^{-4} in. (4×10^{-4} cm.) thick on steel and have explained the action in terms of the theory outlined in the second section of this paper. The same basic principles explain the action of any soft metal used as a lubricant. In practice indium, lead, tin, zinc, copper and barium have all been used for this purpose.

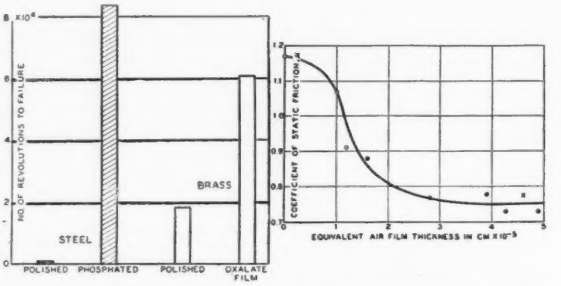

Fig. 6 (left) Wear resistance of a baked resin bonded graphite film on steel and brass surfaces before and after phosphate and oxalate treatment.

Fig. 8 (right) Relation between coefficient of static friction and thickness of tarnish film on copper.

From equation (2) we see that for the friction force to be at a minimum A and s must both be as small as possible. If a soft metal slides on a soft metal, s is low but A, which is determined by the softness of the metal and the load, is high. For a hard on a hard metal, A is now low, but s is high. If the film of a soft metal on a hard base is thin enough, then A will be determined by the flow properties of the hard substrate, and will be small, while s, which will be the shear strength of the soft metal, will also be low.

The mechanism of action is not the same for a film of a soft metal as for a film of an organic solid lubricant in class 2 above, although the sliding characteristics are similar. In the latter case Amontons' law is obeyed, and effective lubrication is obtained for films only one molecular layer (10^{-7} in.) thick. Furthermore, although a thin film of an organic solid can be

worn off the surface by repeated sliding, it is more resistant to this kind of action than a soft metal film, possibly because of higher surface mobility which gives it some measure of self-healing. Soft metal films are not effective in thicknesses below 4×10^{-6} in., and Amontons' law is not obeyed, the coefficient of friction decreasing with increase in load. They provide good friction lowering up to thicknesses of 10^{-3} in. The reason for the satisfactory wearing properties of good bearing alloys such as the babbitts is that during running-in a film of the softer metal in the mix is smeared over the hard, supporting matrix. The surface then behaves as described above. Lead is added to steel to improve its machining characteristics and to bronze to improve its wear characteristics. An interesting example of the use of leaded bronze is in the face plate of the gland closure for the crankshaft in the compressor compartment of refrigerators.

5. Polymer Films: Experience with plastic films as solid lubricants is very limited, but the preliminary results on porous metal impregnated with polytetrafluorethylene (teflon) cited above are very interesting. Shooter's basic studies indicate that the mechanism of friction for plastics is the same as that outlined for metals above, except for teflon which has a very low coefficient of friction [25, 26].

Studies to date on wear of plastics like nylon, and fluorinated polymers indicate that their chief limitation lies in their poor thermal conductivity. Frictional heat is not dissipated fast enough from its source to prevent the local temperature from rising above the melting point of the plastic, and galling ensues. Impregnation of porous metal provides the necessary heat conductivity, but is a diffcult process. The work of Zisman in this issue shows the interesting possibilities of thin fluorinated polymer films.

6. Miscellaneous Soft Solids: A large variety of inorganic solids, which do not clearly fit into any of the classes above comprises this class. Examples are basic lead carbonate or white lead, used in threading compounds, lime, used as a carrier in wire drawing, talc and bentonite, used as fillers in greases for cable pulling.

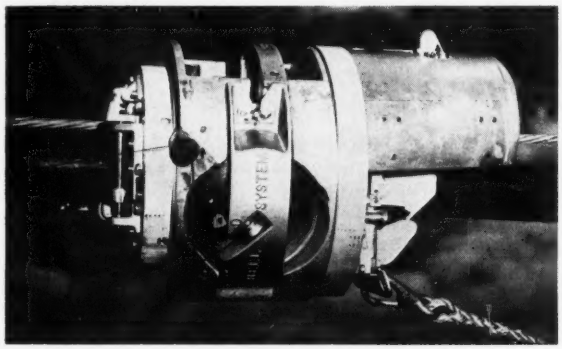

Fig. 7 Solid lubricant on the wire paid out by this lashing machine prevents snarling of the wire.

APPLICATIONS OF SOLID LUBRICANTS: A solid lubricant is only needed where the use of fluid or semi-fluid lubricants is impossible, undesirable or ineffective. It will not in general substitute for a satisfactory fluid or semi-fluid lubricant, although it may supplement it. Although many of the applications have been indicated already, a classification according to type may be of interest.

1) For bearings subjected to a wide temperature range: Lubricants in classes 1, 3, 4 and 6 all qualify for this type of service, but graphite and molybdenum disulfide are the most widely used. All types are suitable for extremely low temperatures.

Air-borne mechanisms pose most of the extreme temperature problems, lubrication of those associated with jet engines and proximity fuses being typical examples. In industry, bearings of trucks or conveyors passing through kilns or furnaces, and of moving parts of ovens, driers and stokers need the help of a solid lubricant.

2) Lubrication of equipment exposed to environments reactive to conventional lubricants: The inertness of layer-lattice lubricants makes them indispensable for many of these applications. Examples are bearings of mechanisms for operating specimen racks in chromium plating, bearings in contact with steam, hot and cold water, gasoline, and other solvents in pumps, meters, mixers and rotary filters.

Fluorinated polymers are exceedingly inert and make useful packing materials for equipment handling highly reactive gases such as fluorine.

3) Lubrication of bearings adjacent to surfaces or materials which must not be contaminated: Many examples of this application are found in electronic instruments, where creep of lubricants onto electrical contacts must be avoided. An interesting example of this class of operation is found when rotation of parts in a vacuum tube is necessary. A barium film on the bearing surface solved a problem of this type[31].

A good example of the importance of avoiding contamination of adjacent materials is found in the pulling of telephone cable into ducts. The lubricant which has been used for years is grease heavily loaded with talc. Shortly after polyethylene cable began to be used, it was found that when stressed it would crack if exposed to soap. Later it was found that almost every organic chemical known would produce the same effect. Since polyethylene and lead cables lie side by side, and some armored polyethylene needs to be lubricated while being drawn into ducts, a substitute lubricant had to be found. A satisfactory job is now being done by a grease compounded from bentonite and water.

Other examples in this class are camera shutter mechanisms, gunsight adjustment mechanisms, textile machinery, door hinges and catches, zippers, window slide surfaces, etc.

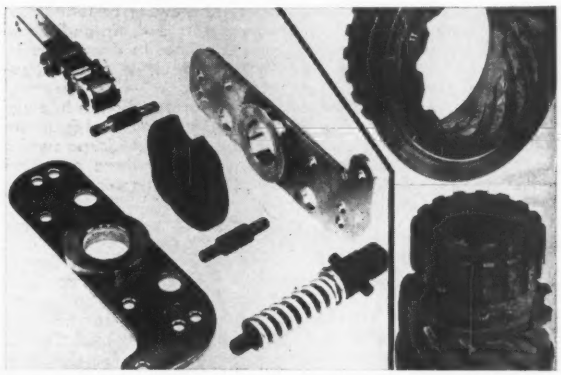

Fig. 9 (left) The stainless-steel cam assembly of the autopilot manual disconnect is lubricated with a bonded, layer-lattice lubricant.

Fig. 10 (right) The helical spline drive and jaw assembly of an aircraft starter are phosphate coated to supplement the action of a low temperature lubricant.

4) Lubrication of mechanisms operating intermittently over long periods or kept in storage for emergency use: Automatic switching devices in communications equipment fall in this class, as do several types of military instruments such as proximity fuses, bomb triggering mechanisms, guided missiles, etc.

The stored military equipment may have a short operating life, so that friction reduction is the only prime necessity, but cleaning and relubrication of parts must be minimized, so as not to contaminate adjacent contacts. The lubricant must be non-gumming, non-dust collecting, and a good corrosion inhibitor. It is difficult to find fluid base lubricants which fulfill all these needs, but several types of solid lubricants fill the bill.

5) Lubrication of inaccessible bearings or those whose design makes lubricant retention difficult: A good example in this class is the mechanism whose parts are assembled in Fig. 9. This is an autopilot disconnect which must operate positively with low friction at all times over a wide temperature range. The stainless steel cam is coated with a bonded layer-lattice lubricant to prevent galling of the trunnion bearings which took place when a conventional synthetic instrument grease was used.

6) Lubrication of Contacts: Organic lubricants cannot be used on contacts making and breaking currents. For this reason, brushes on motors, generators, and on many wire-wound potentiometers are made from compressed graphitized carbon alone or strengthened with various metals.

7) Situations where dust collection is objectionable: In certain textile operations collection of lint in the bearings can lead to stoppage. Solid lubricants are useful in these situations. It is also important to prevent collection of dust on rubbing surfaces of intricate mechanisms which must operate perfectly when required after long periods of storage.

8) Minimization of Fretting: Many classes of solid lubricants — bonded layer-lattice, phosphate, metal, and polymer films have proved their worth[27]. There is room for considerable development in this application.

9) As a supplement to fluid lubricants: Thickeners in greases are an example of this class, although the main use of the thickener is not usually as a lubricant and the oil does most of the lubricating. However, in some applications, notably at high temperatures, the solid lubricant added in high concentrations to temperature resistant fluids like silicones, or polyalkalene glycols, is the primary friction reducing agent[32]. The solids in anti-seize compounds play an essential part in reducing friction and preventing galling. Powdered zinc and lead, graphite, molybdenum disulfide, zinc stearate and white lead are all used in these formulations.

Perhaps the most outstanding example of supplemental action is that afforded by surface conversion coatings such as phosphates. An example of the use of this type of coating is in the lubrication of the helical spline drive and jaw assembly shown in Fig. 10. This is part of an aircraft starter required to operate at very low temperatures. No lubricant giving the right combination of low torque and high lubricity was available and the problem was solved by phosphate-coating the spline drive and jaw and using Mil-L-6085A oil.

10) Metal Working Operations: Every class of solid lubricant is useful in some phase of this field, usually in combination with one or more other solids. Examples are phosphate coatings or lime plus soaps in dry wire drawing or forming, or with emulsions or liquids in wet operation; lead, tin and indium coatings for wire drawing; lacquer as a base in drawing stainless steel tubes; graphite or molybdenum disulfide in wire drawing and forging.

11) Molding Operations: In these operations the lubricant prevents sticking of the molded part to the mold and facilitates relative motion of the particles during shaping. Soaps such as zinc and calcium stearate are widely used in these operations which include plastic molding, pharmaceutical tablet manufacture, powder metallurgy, ceramics.

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Lubrication in the News

FILTER-REGULATOR-LUBRICATOR, a new type containing numerous improvements, features a porous bronze filter element which has the ability to retain on its outer surface a much greater amount of sediment without a serious loss of pressure. Air or filtrate enters the unit where centrifugal action throws the moisture and heavier solids against the bowl and downward into a relatively quiet zone below the baffle where it is deposited. The lubricator has been designed to deliver into the air stream a finely vaporized mist of oil that remains in suspension and provides positive lubrication to the precision working parts of air operated equipment. A porous bronze feeder wick forces the lubricant to enter the air stream in a fine mist. Incorporated in the regulator are several new improvements in design: a new positive action self bleed feature for prevention of pressure build-up in dead end applications, etc., a diaphram of new type oil resistant material with Nylon cord reinforcement, and stainless steel

(Continued on page 225)

THE DEVELOPMENT OF SIGNIFICANT BENCH TESTS FOR AIRCRAFT TURBINE LUBRICANTS*

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The purpose of small scale or bench tests for lubricants is to make possible the selection or screening or approval of oils without the necessity for testing each batch or each shipment of oil in a full scale engine.

In the first place the tests form a part of the oil specification to insure that material sold and delivered will be suitable for use, and in the second place, they provide for screening new oils that are offered so that no time will be wasted in testing poor lubricants in the main engine.

PRESENT POSITION REGARDING TURBINE OILS: When aircraft gas turbines were first put into large production, they were being lubricated with mineral oils which had been developed for other purposes. These early engines operated under what can be called "mild conditions" and fairly satisfactory operation was obtained with ordinary oils. As soon as more experience was gained, however, and the operating temperatures were raised, together with the altitude of operation, the petroleum oils began to show some deficiencies. At the higher temperature and lower ambient pressure, rapid evaporation took place. High bearing temperatures after shut-down caused coking of the oil on the bearing cages. It was apparent, therefore, that better properties were required.

The most obvious reason for the unsuitability of mineral oils for advanced jet engines is the difficulty of getting sufficient spread between the pour point and the boiling point. If an oil is selected which has a low enough pour point then the evaporation loss at working temperatures, especially at high altitudes, is so large as to be prohibitive. As a result, the field of synthetics has been explored for materials which have a big range between freezing and boiling while keeping other desirable properties.

The present situation is that some engines are able to get by fairly well on the straight mineral oil (1010) while the advanced designs which, in general, have higher compression ratio and higher working temperatures, and operate at higher altitudes, require a synthetic material. Even with the older engines, extended flight tests have shown that oil consumption is reduced in a ratio of 24:1 by changing from 1010 to the synthetic material so that it would be economic to use the latter material just on a cost basis.

BENCH TESTS: The suitability of an oil depends largely on its physical and chemical characteristics. For control purposes, i.e., to insure that each successive batch of oil will be suitable, tests are done in some form of bench apparatus. Some of the physical tests are of little value except as they serve the refiner in controlling manufacturing processes while others are useful both to the consumer and the manufacturer as an index of the value or fitness of the product for particular uses. Such chemical tests as now exist serve principally to protect against impurities or undesirable constituents or to assess some particular property such as corrosiveness. Examples of such items are: (1) Viscosity at various temperatures, (2) Viscosity index, (3) Pour point, (4) Low temperature stability, (5) Flash and fire points, (6) Evaporation loss.

It is not enough to hold the physical and chemical properties of an oil within specified limits because this alone does not insure satisfactory performance in the engine. It is necessary to apply some performance tests to the oil. The performance of oil or other material may be studied in several ways. For instance, in the complete engine, in an engine component such as the gear set or the bearings, in apparatus which simulates engine conditions, such as a gear test machine, or in other apparatus which does not resemble any part of the engine. Some factors cannot be satisfactorily tested apart from the complete powerplant. The final answer, therefore, is always obtained in the engine but it is manifestly impossible to do all of our oil testing in full scale engines so that bench tests have a very important place in the scheme of things.

The proper choice of tests and the assignment of proper numerical values to specifications based upon these tests is of funfamental importance. We do not gain merely by including as many tests as possible. It is best to have the smallest number of tests that will still reject all unsuitable material. It is required to determine the effect of a lubricant on the various metals in the lubrication system and the packings and other engine parts and, in addition to this, one must measure the effect of different metals on the lubricant itself.

NEW REQUIREMENTS OF TURBINES: In the change from piston engines to turbines, it was found that some oil requirements remained the same and the old tests could be used. There were, however, new requirements for which no good tests existed and it was necessary to develop such tests. Likewise, in passing from the familiar petroleum oils to new classes of materials with which we had no previous experience, new questions arose.

In considering a new class of materials, there are a number of special tests to be made which will not be incorporated in the specifications nor used as control tests. The toxicity of the material must be investigated. It is also desirable to know the

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specific heat and thermal conductivity as well as the dielectric strength. Some of these properties have a bearing, for example, on the design of such engine parts as oil coolers and oil level gages. Since a good many engine parts are made of rubber or synthetic rubber, it is necessary to know the effect of the proposed lubricant on these materials. Comprehensive bench tests are made to assess rubber swelling and other characteristics but it is not supposed that there will be any great batch to batch variation in these properties so this kind of test will not be used as a routine matter.

DEVELOPMENT OF NEW TESTS: A new test will only be devised when a definite need for it is apparent. This is generally a result of actual trouble experienced in engine operation. One property of the oil must be singled out which is usually connected with the trouble in question. Then a test is devised to emphasize this particular property and to distinguish sharply between lubricants which are good or bad in the final engine operation.

An example of such a test can be taken from piston engine oil experience. Between 10 and 15 years ago there was an outbreak of bearing trouble on engines in airline service. A major improvement in the construction of the bearing was made by the use of the silver-lead-indium combination. As a matter of fact, this bearing is still the prevailing one in piston engines and has made an outstanding performance record. When lubricated with a non-acid oil, it will carry extremely high loads and have long service life. Of course, it is not proof against all kinds of abuse, and lead washing and eventual bearing failure can be caused by the use of oils which become excessively acid in service. To safeguard against this, it was necessary to add to the specification tests a new test for oxidation and corrosion. For this purpose a modified MacCoull test was developed[1]. This test is run in the "10 hole corrosion apparatus" and the modification consists in the use of a lead surfaced specimen instead of MacCoull's copper lead test piece. This test has been eminently successful in keeping us out of bearing corrosion trouble for 12 years and is used both as a screening test for new oils and as a control test for all the engine oil that we purchase.

For turbine oils, either this test or an equivalent one must be used to insure against fast oxidation and deterioration of the oil at working temperatures. There are other corrodible materials in the engine beside lead and our present practice is to use more than one kind of corrosion test, with several metals present.

THE COKING TEST: Another test which carries over from the piston engine regime is the coking test. This was originally developed on account of rocker box coking. It is not that this type of carbon formation is so harmful in itself but it is a symptom of breakdown and deterioration of the oil. The rocker box is a convenient place to look for it because that is one of the hottest spots touched by the oil in its circulation through the engine. This is not the first time that someone has splashed oil on a hot plate to see what happens but the coking apparatus is now developed to a state where reproducible results are obtained and we think it is doing a good job of keeping us out of trouble on this particular front. In the turbine engine we do not have rocker boxes to fill up with coke but there may be heavy carbon formation on the bearing cages and, if this proceeds very far, the balls or rollers are prevented from turning freely and slippage occurs, with rapid wear. In the earlier engines the highest bearing temperatures occur after the engine has been stopped. In later engines, however, the working temperatures are higher and there is very little increase after stopping. In any case, the bearing nearest the turbine wheel runs at a higher temperature than we have been accustomed to in older practice and we must have a lubricant that will stand this high temperature without excessive breakdown.

The idea for the coking test rig came from the California Research Corporation[2]. Some minor modifications were incorporated as use of the apparatus showed the need for them. For instance the support of the hot test plate had to be changed to prevent excessive coking around the edges. The oil bath level was maintained by the addition of a chicken fountain. This apparatus is now made in a standard form by a commercial toolmaker and is available to anyone.

THE GEAR LUBE TESTER: Development of the coking apparatus turned out to be easy and quick but development of a gear test rig has consumed many years and a great deal of expensive experimental work.

In 1941 we decided to build a bench machine which could be used both for the testing of gear materials and gear lubricants. This was to use actual gears as test specimens since no other bench tests seemed to show correlation with actual results in full scale gear sets or geared engines. A couple of years were spent in building the first machine and running it long enough to find out what was not right. The apparatus was then completely redesigned and a second machine built. This eliminated many of the defects of the earlier one and was used for extensive fatigue tests of gear materials.

When the turbine engine came on the scene with its requirement for low viscosity oils in order to facilitate starting at low temperatures, it was evident that gear lubrication would be a problem. It was desirable to use the same lubricants for turbojets and turbo-props unless it should be found that this was clearly impossible. Most of the work on this subject would be done on the gear rig since we already had considerable correlation with engine operation and, in any case, new engine tests would be made from time to time on selected lubricants. It was soon found that although the light mineral oil (1010) had very little ability to lubricate gears at 10,000 RPM (0-600 PPI) it was easily fortified with chemicals to make it suitable for this purpose. The same was found to apply to the various synthetic materials that were offered from time to time as improved turbine lubricants. As we already had the gear tester available and the test procedure well worked out, we saved an enormous amount of time and money in this investigation of gear lubricants. The gear test machine has been described elsewhere[3].

We have not measured the long-time wear of gears as a function of the lubricant but some recent work by others, for example, Hutt[4] indicates that the fatigue life is somewhat less as viscosity is reduced. This work also shows very defnitely that the corrosive types of EP agents cause a marked re-

duction in fatigue life of the gears, doubtless due to corrosion fatigue or something equivalent to it. This is one of several reasons why corrosion cannot be tolerated and it is necessary for us to have adequate corrosion tests for the oil.

THE QUESTION OF CORRELATION: For aircraft engine gears a design Hertz stress of 125,000 to 140,000 psi is permissible. In the gear test machine, a Hertz stress of 125,000 psi is obtained at an indicated load of 1050 lbs. per inch width (PPI) of tooth. 140,000 psi Hertz corresponds to 1340 ppi. We can therefore conclude that scuffing values should be greater than 1050 to 1340 in order for the oil to permit the gears to run long enough to fail by fatigue (pitting).

In actual engine tests, the gears have been examined after running with each of several lubricants and we find the following situations: Oils which test 200 to 700 (PPI) in the gear rig are unsuitable for use in the main engine. Gears immediately show signs of scuffing if not actual scuffing and scoring. Oils which test 1000 to 1400 (PPI) can be called borderline. There may or may not be trouble but it is indicated that in the long run trouble would result. Oils testing 1700 to 2250 (PPI) and upward have given no gear troubles in the main engine and while we have never had time and resources to make main engine tests of gear endurance, we are confident that lubricants which test in this region are perfectly adequate to lubricate at loads high enough to ultimately produce fatigue failures. Of course, the design loads in approved engines are somewhat less than this.

Some samples of lubricants have shown very high anti-scuffing properties and if their other properties can be made entirely suitable then the minimum scuff load can be increased. The values quoted above have been obtained on the Pratt & Whitney Gear and Lubricant Tester, Full-Scale Reduction Gear Sets and Pratt & Whitney complete engines.

The Four-Ball apparatus has been extensively used by various laboratories for testing the so-called film strength of lubricants. We have used it ourselves but I must confess that we do not have any satisfactory correlation between Four-Ball results and main engine results. As far as I know, this same statement holds true for other engine builders and oil laboratories.

A RATING SYSTEM FOR OILS: When a specification merely defines the minimum acceptable quality of a product, without distinguishing between various degrees of goodness or acceptability, competition, if nothing else, influences the suppliers to work close to the minimum and compete solely on a price basis. However, if, as is true with many products, oils can be graded according to their utility, the producer should be able to command a higher price for a superior material.

As an example, coal is sold on a BTU basis. Gasoline is all pretty much alike except for octane number, and is priced both at wholesale and retail on an octane number basis. Eggs command premiums for size, color, etc. This general principle is widely accepted.

In the case of piston engine oil (for aircraft) there has been no major improvement made in the last 13 years. Many tests have been made, using field equipment in every-day use, in an effort to demonstrate that one or another brand of oil would show superior performance. To the disappointment of all concerned, no great difference in performance has been shown. With the coming of the turbine engine, however, necessity has thrown open the door for the use of many new classes of materials including non-petroleum liquids, and our specifications must be drawn so as to encourage the search for better and better properties.

I have already discussed the importance of both the low and high temperature properties of turbine oils. There are engines which operate at mild conditions and those which are more severe. Moreover, operating conditions will sometimes dictate readiness to cope with extreme cold, while other situations may never require this. As to other properties, we might not tolerate different degrees of badness; for instance, we would not like to have two grades of oil which differed only in corrosiveness. Resistance to temperature effects seems to be the thing to use to sort out usable lubricants into classes which will be suitable for different classes of engines, and will presumably be sold at different prices, depending on the duty involved.

It is proposed by E. A. Droegemueller that a dual numbering system be adopted for oils, in much the same way as performance numbers are used for aviation gasoline. The first number will indicate the ability to function at low temperature; the temperature at which the viscosity is 5000 cs might be a good number. The second number will show the upper limiting temperature; this might perhaps be the temperature which will produce 100 mg. of coke in the standardized coking apparatus in an 8 hr. test. The coking test will be set up in such a way as to eliminate oils showing excessive evaporation losses at the test temperature (although it may still be desirable to retain the regular evaporation test as a more accurate measurement. Perhaps the result of the latter would be reported, but not limited by the spec.)

As an example, a current brand of 1010 oil is rated by this scheme as —40/400. This indicates at once that the oil is suitable only for mild conditions, but may be entirely adequate for starting above —40 F., and for operation where the lubricated engine parts will not get above 400 F.

Since the oil consumption of turbine engines is very low, the price per gallon is not going to be a very large factor in the selection of oil. At any rate, the situation should be much better than it has been in the case of fuels and price should not be allowed to forbid the purchase of oil which will be fully adequate for the service intended. A limited number of specification grades will be called for but these specifications will merely set floors under the important properties. Within the separate grades, suppliers can compete with each other by offering a little more on the cold end or a little more on the hot end.

CONCLUSIONS: Among small scale tests, performance tests are seen to be just as important as the more conventional chemical and physical tests. A useful performance test is only adopted because of actual or potential trouble on the operating side. It must therefore measure some quality which is capable of being changed in such a way as to cause trouble in the engine. Out of the hundreds of possible tests, a very few are selected which have real significance because they bear a real relation to actual full scale performance. A rating scheme is

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FRICTIONAL BEHAVIOR OF POLYETHYLENE, POLYTETRAFLUOROETHYLENE & HALOGENATED DERIVATIVES *

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INTRODUCTION: The increased use of synthetic high polymers as protective coatings, dielectric materials, and fibers has created interest in their mechanical and frictional properties. Our own research has been especially concerned with two aspects: (a) the effect of chemical constitution on the frictional properties of linear high polymers made from halogenated derivatives of ethylene, and (b) the properties and applications of such polymers when firmly bonded to hard-metal substrates.

Within a few years after the discovery of polytetrafluoroethylene (TFE) by Plunkett[1], our observation of the uniquely low coefficient of friction of thin sheets of pressed TFE prompted us to call to the attention of Dr. W. S. Calcott of the DuPont Company their possible use as dry bearing materials. This led the Organic Chemicals Department of that organization to experiment with new bearing materials made by sintering together compressed powders of TFE and of metals. These were found to exhibit excessive plastic flow at high unit loads, the effect being accelerated by localized heating of the rubbing surfaces and poor heat transfer through the composite bearing material[2]. It was finally decided that these bearings would be restricted to very special uses, such as in oilless bearings for small instruments and motors operating under low unit loads and at low speeds. One of the most interesting of the many products developed from TFE in the postwar years has been the aqueous dispersions of polytetrafluoroethylene for use in coating solids with a thin film of TFE[3, 4, 5, 6]. TFE suspensoids of sufficient stability and uniformity are commercially available and are being widely applied. This development has permitted us to extend our wartime interest in the possibility of putting thin films of TFE to work as solid lubricating coatings.

During the past ten years, various linear polymers of other halogen substituted ethylenes have been prepared, many of which have been made available in research quantities for our studies. This led to an investigation of the effect on the wettability[7, 8, 9, 10] and on the frictional properties[11] of replacing the hydrogen atoms in polyethylene by either fluorine or chlorine atoms. This report presents a full account of the experimental methods used, the frictional properties observed, and recent applications to lubrication of the friction reducing and wear preventing properties of some of these plastics.

EXPERIMENTAL METHODS & MATERIALS: The frictional force between a ball slider and a plate of the plastic under study was measured with a modification of the well known Bowden-Leben "stick-slip" machine[12]. Our machine was built around a Brown and Sharpe No. 5 surface grinder having both the cross feed and longitudinal movement of the table driven hydraulically. The hydraulic pump had been removed from the interior of the machine to a distance of 15 feet so that pump vibrations could be sufficiently damped through the long pressure lines. A plate-shaped specimen or platen is clamped to a table, which is hydraulically driven at a uniform velocity. A hemispherically ended slider is attached to a flat steel spring which is secured to a bifilar suspension. The two specimens are pressed together with a known force by utilizing the restoring force in the flat steel spring. A light beam is reflected from a mirror which is also attached to the flat steel spring, and it is focused on a modified oscilloscope-type magazine camera which is located 675 cm from the mirror. A tangential force of 150 grams will deflect the light beam one centimeter on the camera. Frictional measurements were made between combinations of steel-on-plastic, plastic-on-steel, and plastic-on-plastic with a constant sliding velocity of 0.01 cm/sec and at a temperature of 25 C.

Two types of polymer surfaces were used—abraded and heatcast. The first type was prepared by abrading the plastic under water in a rotary manner with 600A grit "Wetordry Waterproof Tri-m-ite" silicon carbide paper, and the second type of surface was prepared by pressing the plastic for a short time against a highly polished, flat, nickel disc held at a temperature above the polymer softening point. While being quickly cooled to 25 C., the metal disc and plastic were kept under an external pressure of 2000 psi. Examination of these surfaces with a Short Sclerscope showed that they were softer than the abraded surfaces—because of the quick cooling they may have been amorphous.

Polytetrafluoroethylene (TFE), polytrifluorochloroethylene (TFCE), and their copolymers were cleaned in a boiling (1:2) nitric-sulphuric acid mixture, then rinsed in three successive changes of boiling distilled water[7, 8] and dried in a desiccator. They were also cleaned by abrasion under water with 600A grit "Wetordry" paper, rinsed in distilled water, and dried as before. In a third method the material, after abrasion under water, was cleaned with a concentrated solution of a detergent (Tide), then washed with plenty of distilled water, and finally dried in a desiccator. The same frictional measurements resulted after each method of cleaning. Only the last two methods could be used for cleaning these plastics subject to attack by oxidizing acids. Identical frictional measurements resulted from using these two methods. Therefore, in the majority of the measurements made the third method of cleaning was used.

Flat steel plates or platens and spherical steel sliders were all made from low-carbon steel hardened to Rockwell C-10-12, except those 6.35 mm and 12.7 mm in diameter, which were commercial balls hardened to Rockwell C-62-63. The platens were cleaned by lapping each under water with 600A grit "Wetordry" paper and wiped on grease-free filter paper to remove any grit, washed with a detergent (Tide) to remove any cellulose fibers, rinsed with freshly distilled water, and finally dried with C.P. acetone. The steel sliders were boiled in C.P. benzene; and, just prior to being used, were polished with Behr-Manning 4/0 emery polishing paper.

Chemical names, structural formulae, sources, molding conditions, and some pertinent physical properties of the polymers of interest are listed in Table I. The compositions of the series of copolymers made of chlorotrifluoroethylene monomers and tetrafluoroethylene monomers[13] as well as the copolymer of the latter with ethylene[14] are expressed in mole percent of the two monomers. These copolymers were available only as thin translucent sheets approximately 0.050" thick. The 50-50 copolymer of tetrafluoroethylene and ethylene was molded as flat plates ⅛" thick. Polyvinyl fluoride is a rare material and was obtainable only as a thin translucent sheet approximately 0.0025" thick. Because of the dimensions and forms of the above plastics, the study of their frictional properties was limited to that of steel sliding on the plastic. Specimens of unplasticized polyvinyl chloride, polyvinylidene chloride, and polyethylene were carefully molded by us from their powders as transparent and nearly colorless discs 2.8 cm in diameter and 0.5 cm thick and as hemispherical specimens ½" in diameter. Test specimens of TFCE and TFE were made from bulk plastic in the shape of flat plates and hemispheres by milling or by turning in a lathe with a grease-free tool.

Thin adherent films of TFE were prepared both on a steel and on a brass plate by using commercial TFE suspensoid in water. A freshly cleaned steel surface was coated with the suspension by spraying or dipping, was allowed to dry, and then was fused either for 3 to 4 minutes at 700 F. in an electric oven or for a few seconds in a radio-frequency induction heater with a frequency of 400 kilocycles. No difficulty was encountered in preparing remarkably smooth and uniform films. The thickness of the film deposited could be decreased by diluting the suspensoid with distilled water, and film thicknesses of 0.0005" to a tenth of that were readily prepared.

GENERAL RESULTS: It is well known that the Bowden-

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Leben friction and wear machine will develop a characteristic relaxation oscillation or "stick-slip" motion when the coefficient of kinetic friction (μ_k) is much less than the coefficient of static friction (μ_s). Incidence of "stick-slip" is dependent on the friction-velocity relation as well as on the natural frequency, inertia, and damping of the vibrating parts of the machine[15, 16, 12]. For a given coefficient of friction, the amplitude of "stick-slip" vibration increases with the load applied to the rubbing surfaces, while for a given load, the amplitude increases with the coefficient of friction. Bowden and Tabor[12] have shown that rubbing surfaces having large coefficients of friction are damaged or worn largely because the adhesion of contacting asperities of the two surfaces is followed by their tearing or shearing. Under extreme conditions of surface damage, these two processes may coincide with the incidence of the "stick" and of the "slip", respectively. Therefore it is usual to supplement the data obtained by this machine with photomicrographs of the damaged surface of the ball and of the wear path after a single traverse.

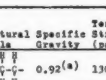
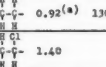
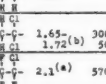
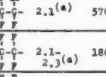
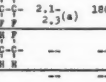
Polymer (Unplasticized)	Structural Formula	Specific Gravity	Tensile Strength (psi)	Elongation Percent	Modulus of Elasticity in Tension (10^5 psi)	Hardness (c)	Molding Temp. °F.	Molding Pressure (psi)
Polyethylene[1] M.W. 25,000		0.92[a]	1300[a]	200[a]	0.19	46	250-275	2000
Polyvinyl Chloride[2] M.W. 85,000		1.40				71	300-325	2000
Polyvinylidene[3] Chloride		1.65-1.72[b]	3000-5000[b]	100-200[b]	0.5-0.8[b]	64	250	2000
Polytrifluorochloroethylene (TFCE)[4]		2.1[a]	5700[a]	28-36[a]	1.9[a]	77	450	2000
Polytetrafluoroethylene (TFE)[5]		2.1-2.3[a]	1800[a]	110[a]	0.58[a]	30		
Polyvinyl Fluoride		--	--	--	--	--	--	--
TFE - TFCE Co-polymers[5] Mole Percent TFE / TFCE: 98 / 2; 90 / 10; 80 / 20; 60 / 40; 10 / 90		--	--	--	--	--	--	--
50-50 Mole Percent[5] Copolymer TFE-Polyethylene M.W. 25,000		--	--	--	--	--	--	--

Table I Structural formulas & physical properties of the high polymers. Key: (1) Bakelite Div., Union Carbide & Carbon Corp. (2) The B. F. Goodrich Chemical Co., a Division of the B. F. Goodrich Co. (3) The Dow Chemical Co. (4) Chemical Mfg. Div., The M. W. Kellogg Co. (5) E. I. duPont de Nemours & Co., Polychemicals Dept. (a) Plastics Properties Chart: Modern Plastics Encyclopedia & Engineer's Handbook 1951. (b) Dow Chemical Co.: Modern Plastics Encyclopedia 1949. (c) Shore Scleroscope Hardness (for abraded surfaces).

Although the lower specimen may be kept moving at a constant velocity with respect to a fixed point on the apparatus, the relative velocity between the two rubbing surfaces becomes zero during the "stick". Thus, a series of static values may be measured during sliding. In Table II the initial value of μ_s and also its value measured after the ball has traveled 2 cm are given for each plastic exhibiting "stick-slip" motion. Of course, only one value of μ_s can be reported for those plastics which slide freely without "stick-slip".

Measured coefficients of friction are reported in Tables II and III for both abraded and polished surfaces. It was impractical to abrade or to polish further the thin sheets of fluorinated copolymers; hence, the results reported are for those surfaces as prepared by the manufacturer. In the first group of Table II are found a series of partially chlorinated polymers of ethylene, which are listed in order of increasing chlorine substitution. In the second group are the results for polyethylene, polyvinyl fluoride, the 50-50 copolymer of tetrafluoroethylene and ethylene, and finally TFE. Variations in the kinetic coefficient of friction (μ_k) with distance of travel are indicated by the range of values given in Table III for the polymers which did not show intermittent motion. Thus, μ_k for polyethylene decreased from a maximum value of 0.33 to a constant value of 0.25 when steel slid on the abraded polymer, and it decreased from a maximum value of 0.60 to a constant value of 0.50 on the polished polymer.

Specimens of TFE, polyethylene, TFCE, polyvinyl chloride, and polyvinylidene chloride were each subjected to a series of friction measurements in which the ball diameter was varied between 6.35 and 50.7 mm, and the load was varied from 0.2 to 1.5 kg. It was found that for each polymer F/W (or the ratio of the frictional force F and the load W) was constant. Hence, Amonton's law is obeyed over the load range employed, so that the coefficient of friction (μ) is independent of the load. In Tables II and III the values of μ given were obtained with balls 12.7 mm in diameter and a load of 0.8 kg.

Friction measurements on polyvinyl fluoride could be made only for steel sliding on plastic. It was observed that μ_k decreased from 0.30 to the constant value of 0.10. For steel sliding on TFE, μ_k decreased from 0.10 to approximately 0.04 with an abraded surface and from 0.15 to 0.08 with a polished surface. TFE sliding on steel exhibited a decrease in μ_k from 0.10 to a constant value of 0.04. It remained constant at 0.04 as abraded TFE slid against abraded TFE. The coefficients of friction of steel sliding on the abraded 50-50 copolymer of TFE and polyethylene were nearly identical with that found for polyethylene, i.e., μ_k decreased from 0.33 to approximately 0.25.

In Fig. 1, μ_s and μ_k for steel sliding on each of the series of TFE-TFCE copolymers are plotted against the percent of chlorine atoms replacing fluorine. Both coefficients of friction rise with the chlorine content, and especially with the first 10 percent. A nearly constant value of μ_k is reached after the steel ball has traversed a distance of 1 cm, and it is this value of μ_k which is plotted on the lower curve. "Stick-slip" motion did not occur on copolymers containing 10 percent of chlorine, but did occur with 20 percent. Consequently, the value of μ_k shown for the latter materials was estimated in the usual way as being one-half the sum of the "stick" and the "slip" values.

TABLE II

Polymer	Steel Sliding on Polymer: Polished	Steel Sliding on Polymer: Abraded	Polymer Sliding on Steel: Polished	Polymer Sliding on Steel: Abraded	Polymer Sliding on Polymer: Polished	Polymer Sliding on Polymer: Abraded
Polyethylene	0.60	0.33	0.60	0.33	0.60	0.33
Polyvinyl chloride	0.90 - 0.70*	0.45*	0.45 - 0.60*	0.45 - 0.50*	0.60 - 0.90*	0.45 - 0.55*
Polyvinylidene chloride	1.8 - 1.0*	0.68*	1.4 - 1.6*	0.70 - 0.95*	1.3 - 2.0*	0.80 - 0.95*
Polyethylene	0.60	0.33	0.60	0.33	0.60	0.33
Polyvinyl fluoride	0.30	--	--	--	--	--
50-50 Copolymer of TFE Polyethylene	--	0.33	--	--	--	--
Polytetrafluoroethylene	0.15	0.10	--	0.10	--	0.04

TABLE III

Polymer	Steel Sliding on Polymer: Polished	Steel Sliding on Polymer: Abraded	Polymer Sliding on Steel: Polished	Polymer Sliding on Steel: Abraded	Polymer Sliding on Polymer: Polished	Polymer Sliding on Polymer: Abraded
Polyethylene	0.60 - 0.50	0.33 - 0.25	.60	.33	.60	.33
Polyvinyl fluoride	0.30 - 0.10	--	--	--	--	--
50-50 Copolymer TFE-Polyethylene	--	0.33 - 0.25	--	--	--	--
TFE	0.15 - 0.08	0.10 - 0.04	--	0.10 - 0.04	--	.04

TABLE IV

Polymer	p_2 Kg/mm²	p_2 Kg/mm²	s/p	μ_s
TFE	1.68	4.0	0.42	0.10
Polyethylene	0.91	2.97	0.31	0.33
TFCE	3.32	15.9	0.21	0.43
Polyvinyl chloride	5.05	14.9	0.34	0.45
Polyvinylidene chloride	4.06	6.67	0.61	0.68

TABLE V

Polymer	(μ_s) Steel Ball Sliding on Coated Steel: Calculated	Measured
TFE	.01	.04
Polyethylene	.03 to .04	.05
Polyvinyl chloride	.24	.30
TFCE	.24	.16
Polyvinylidene chloride	.16	.25

Table II (top) Static coefficient of friction (μ_s) of some high polymers (measured at 0.8 kg. normal load and with a 12.7 mm diameter slider). *: "stick-slips" occur.

Table III (second from top) Kinetic coefficient of friction (μ_k) of high polymers free from "stick-slip" behavior.

Table IV (second from bottom) Comparison of shear strengths & yield pressures of the polymers with the coefficients of friction.

Table V (bottom) Coefficients of static friction of thin coatings of polymers on steel.

FRICTION OF STEEL SLIDING ON CLASS I PLASTICS: It is convenient in presenting the remaining data to divide the plastics studied into two classes: Class I, which is made up of those that did not "stick-slip"; and Class II, all the other plastics.

During the sliding of the steel on the Class I plastics, μ_k always decreased attaining a constant value after traversing approximately one or two centimeters. A typical example is that of steel sliding on abraded polyethylene (Fig. 2). The most evident explanations are that either the real area of contact was decreasing, or else a film of the polymer was being transferred to the sliding ball. The first explanation is not plausible, because the width of the track did not vary along the track length. In another experiment, after the first traverse of the ball on the plastic, the ball slider without being cleaned was promptly made to traverse a new area of the plastic, and μ_k was found to remain the same as the value of 0.25 found at the end of the first traverse. This would be expected if a film of polyethylene were transferred to the steel ball during the first traverse. If this is so, it would be expected that the behavior of the sliding combination in the second traverse would simulate that of polyethylene sliding on polyethylene. But when bulk polymer was rubbed on bulk polymer, μ_k was 0.33 instead of

0.25. From this it would appear that the frictional properties of the transferred film differed in some way from those of the bulk material; presumably, the transferred film was oriented. If this is true, μ_k must be lowered by 0.08 when the polymer molecules become oriented in the plane of sliding.

Similar experiments were conducted with a steel ball sliding in one direction on an abraded TFE platen, and the results were similar to those obtained with polyethylene. The value of μ_k decreased during only the first 0.1 cm of traversed path to a constant value of 0.04, which equalled μ_k for the bulk plastic. When a second traverse was made on a clean area without cleaning the slider, μ_k remained constant at 0.04; hence transfer of TFE to the steel ball had occurred in the first traverse. These experiments could not similarly give information on the orientation of the TFE coating on the ball because here the observed constant value of 0.04 also happens to be identical with the value of μ_k for bulk TFE sliding on bulk TFE.

Coefficients of static friction were always much higher when plastics of Class I were "heat polished", the greater friction being attributed to the amorphous nature of the surfaces and to the increase in the real area of contact caused by the accompanying softness of the surfaces. During sliding a general decrease in μ_k occurred much the same as that observed with the abraded plastics.

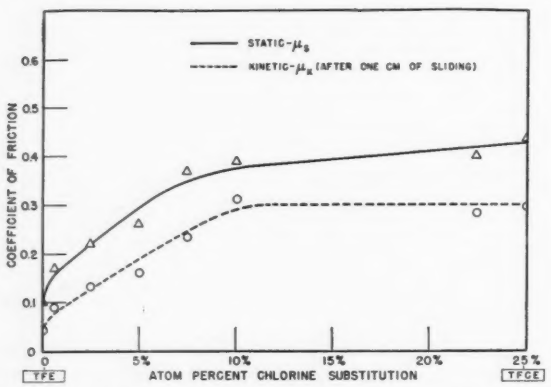


Fig. 1 Coefficient of friction for steel on TFE—TFCE abraded copolymers.

FRICTION OF STEEL SLIDING ON CLASS II PLASTICS: As steel slid on the surface of any of the Class II polymers (i.e., polyvinyl chloride, polyvinylidene chloride, and TFCE), μ_s remained nearly constant for each abraded material, but there was a marked decrease in μ_s with sliding distance for heat polished surfaces. In Fig. 3 are plotted the coefficients of friction for a steel ball sliding on polished TFCE. Most pronounced of all was the decrease of μ_s in sliding steel on polished polyvinylidene chloride. In this instance the initial value of μ_s was 1.8 and the value at the end of the traverse was 1.0. Observation of the wear track revealed that its width decreased in the direction of sliding. Immediately after this experiment the frictional force was remeasured on a clean area of the platen but without recleaning the steel ball, and μ_s was found to have become constant at approximately 1.0. In contrast to the first traverse, a very uniform and narrow track was observed. It is concluded that the rubbing surface of the steel ball had been coated with polyvinylidene chloride. Presumably, the lowered value of μ_s is due to the orientation of the transferred polymer.

It was observed that at the beginning of the track of the first traverse there appeared to be a series of circular indentations which became less pronounced as sliding proceeded. These may be explained by a consideration of the mechanics of the elastic measuring system. Here, μ_s is so great that the bifilar suspension is strained beyond its vertical axis of rotation, the axis of the ball slider is rotated beyond its normal position with respect to the platen, and the spring-steel band pressing the surfaces together is twisted. This results in the ball slider making a series of dynamic type indentations when slip occurs. Presumably, as the film transferred to the slider becomes oriented, the friction decreases and the mechanical system becomes less strained, thus reducing the impact of the dynamic indentation. The smaller area of contact which results, together with the oriented film, reduces the adhesive force.

FRICTION OF PLASTIC SLIDING ON PLASTIC: Of the Class I plastics only TFE and polyethylene were available in a shape convenient for measuring the friction of the bulk plastic against itself. Abraded surfaces of each plastic were used, and it was found that μ_k exhibited constant values during sliding of 0.04 and 0.33, respectively. Since μ_k remained constant during sliding, it is presumed that no orientation of the polymers occurred. When the surfaces of the polyethylene specimens were heat-polished, μ_s and μ_k approximately doubled. The greater friction is attributed to the amorphous nature of the surface and to the increase in the real area of contact caused by its softness.

Class II plastics (TFCE, polyvinyl chloride, and polyvinylidene chloride) all exhibited intense localized adhesion and shearing. With the exception of TFCE, μ_s increased as sliding proceeded (see Table II); and this increase was much more apparent when the polymer surfaces were heat-polished. Examination of the wear track made by polished polyvinylidene chloride sliding on itself showed that its width also increased as sliding proceeded. This could account for the increase in μ_s. Photomicrographs of portions of the wear track and slider wear-scar made evident the large and intermittent transfer of material by localized adhesion and shearing during the relaxation oscillation.

FRICTION OF PLASTIC SLIDING ON STEEL: TFE and polyethylene were the only Class I plastics that could be studied conveniently by sliding a plastic hemisphere on a steel platen. Using an abraded TFE slider, μ_k rapidly decreased during one traverse from 0.10 to 0.04. At the moment sliding started, the value of μ_s represented the friction of polymer sliding on clean steel. At this point, it should therefore equal the static coefficient of friction for the reverse sliding condition, i.e. for clean steel sliding on the polymer. The data of Table II support this conclusion. Promptly after relative motion began TFE was presumably transferred to the steel platen, and the sliding combination was essentially polysliding on polymer coated steel. Therefore, it is not surprising that μ_k for TFE on steel dropped sharply from 0.10 to 0.04 as sliding began, because μ_k for bulk TFE sliding on itself is 0.04.

An abraded polyethylene slider traversing a clean steel platen had a value of μ_k of 0.33, the same as for bulk polythylene sliding on itself. Here one would expect μ_k to have decreased to the same value of 0.25 found earlier at the end of the traverse of a steel ball over a polyethylene platen. But it is to be noted that as a polyethylene hemisphere slides on clean steel, material of the slider near the leading edge adheres to the steel platen, is sheared off, and then is rubbed into an oriented condition as the rest of the slider moves over it. Hence, the observed coefficient of friction of 0.33 is not entirely that of the slider against a transferred film of oriented polymer. However, as a steel hemisphere starts to slide over a polyethylene platen, some polymer adheres to the spherical steel surface, and it remains there to be rubbed against the polyethylene platen during the rest of the traverse. Hence, it is oriented more and more during the motion, so that near the end of the traverse, the coefficient of friction of 0.25 must be that of oriented polyethylene on nearly unoriented polyethylene. When the polyethylene ball was heat-polished both μ_s and μ_k were greater. The fact that μ_k remained constant may be explained in the same manner as above.

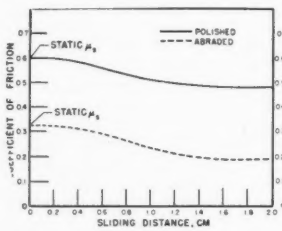


Fig. 2 (left) Coefficient of friction, steel on polyethylene.

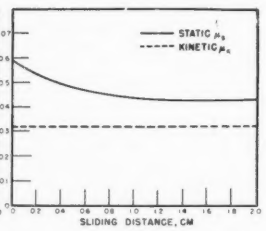


Fig. 3 (right) Coefficient of friction, steel on polished TFCE.

A steel platen and a steel ball 12.7 mm in diameter were both rubbed in one direction with bulk polyethylene, and then the coefficient of friction was determined between the coated metals at a normal load of 0.8 kg. A very low and constant kinetic coefficient of friction of 0.05 resulted. Microscopic examination of the above mentioned steel platen revealed that small particles of polyethylene were scattered widely over the rubbed area, although there was no visual evidence of a continuous film. The friction measurements, however, showed that polyethylene must have formed a nearly continuous film when rubbed on the steel surface, for the steel was so well covered as to prevent metal-to-metal contact and to result in a constant value of μ_k.

A steel ball and a steel platen were each rubbed with

TFE in a manner similar to that previously used with polyethylene, but μ_k was found to vary erratically between 0.25 and 0.10 during sliding. Microscopic examination revealed some large and a great many small particles of TFE scattered over the surface of the steel. It is concluded that the transferred film of TFE is spotty rather than continuous, varies greatly in thickness, and is definitely less uniform and thinner than the film of polyethylene prepared in the same way. Wherever the film was interrupted, the steel slider made contact with the hard metal substrate and the coefficient of friction increased suddenly. This was the cause of the observed erratic variations of μ_k between values of 0.25 and 0.10.

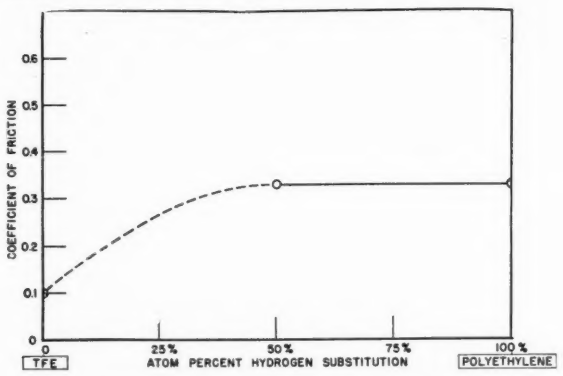


Fig. 4 Effect of hydrogen subsitution on static coefficient of friction of TFE (steel sliding on abraded plastic).

Electron diffraction patterns were obtained by the reflection method with a G. E. Diffraction Unit Model G-2 at a potential of 30,000 volts. A series of good patterns was obtained on examining a steel platen which had been rubbed in one traverse by an abraded polyethylene hemispherical slider. These patterns established beyond doubt that polyethylene had been transferred from the slider to the steel platen, and that the crystallites of this material had become well oriented so that the long axes of the polymer molecules were in the plane of the platen and were nearly all pointed in the direction of sliding. This orientation is like that indicated by the change in μ_k from 0.33 to 0.25 during successive traverses of steel on a polyethylene platen. Hence, in each case the polyethylene transferred to the steel was oriented. A similar electron diffraction examination of a steel platen rubbed by an abraded TFE slider showed that the polytetrafluoroethylene molecules were also highly oriented in the surface of the platen and were pointing with the long axes in the direction of sliding.

It is difficult to see how such pronounced orientation could have developed without involving much slipping either at the metal-plastic interface or between the surface of the bulk plastic and the exposed surface of the transferred plastic. The latter is the more likely for polyethylene sliding on steel.

When the abraded polymers of Class II slid on steel, μ_s increased as sliding proceeded (see Table II). Polyvinylidene chloride exhibited the greatest increase, polyvinyl chloride showed a small but detectable increase, and the increase with TFCE was very slight. The increase in μ_s during the traverse may have been due either to the softening of the polymer by frictional heat generated during the "slips" or to the retarded elastic nature of the plastic. Since the adhesion and subsequent velocity of slip are greater for polyvinylidene chloride, it is reasonable to assume that there will be greater softening at the points of contact than for polyvinyl chloride. A larger increase in the slider area of contact would result for polyvinylidene choride as sliding proceeded and this would cause a greater increase in μ_s. TFCE has a much higher molding temperature than the other two polymers, and the adhesion and slips are smaller; hence, there is little increase in the apparent area of contact or in μ_s as sliding proceeds. In the beginning of the traverse, Class II plastics exhibit higher coefficients of friction when polished than when abraded, and the same type of increase of μ_s as sliding proceeds. But as the soft polished surface was worn away, the frictional force began to decrease and eventually approached the value characteristic of an abraded surface. A polished polyvinylidene chloride ball approached this condition after sliding only a few centimeters. A photomicrograph revealed that the amount of polyvinylidene chloride adhering to the steel platen decreased as sliding proceeded.

FRICTIONAL PROPERTIES OF TFE FILMS ON METALS: Studies were also made of smooth, uniform, thin films of TFE deposited on steel and brass using a commercial TFE suspensoid in water. With ball sliders 12.7 mm in diameter and a normal load of 0.8 kg at 78 F., the value of μ_s observed was 0.04, and μ_k was constant and the same in value. Increasing the temperature to 250 F. had no effect upon either μ_s of μ_k. It will be noted that the value of 0.04 for the initial coefficient of static friction is lower than that of 0.10 reported in Table II for steel sliding on abraded bulk TFE. But 0.04 is precisely the value of μ_s for TFE sliding on TFE; hence, it is concluded that TFE must have been immediately transferred to the slider. Apparently, the physical nature of the baked-on TFE coating is such that the transfer of particles of polymer occurs immediately upon loading the ball; and it appears impossible to determine μ_s for uncontaminated steel sliding on this type of coating. The ease with which TFE particles from the coating are transferred during sliding is presumed to be due to the fact that the particles in the coating are not sintered at all points of contact; in bulk TFE, which is made by fusion and high compression, the particles are probably sintered at many more points of contact.

Because of the ease with which TFE particles from the coating were transferred by rubbing against the ball slider, the durability of the baked-on coating was examined. Static and kinetic coefficients of friction were measured at regular intervals as a steel slider was moved 100 times back and forth at the rate of 0.1 cm/sec. over the same path, the slider always being removed and cleaned just before each friction measurement. After twenty traverses over the same track there was no change in the coefficient of friction ($\mu_s = \mu_k = 0.04$). After thirty traverses both μ_s and μ_k increased slightly. Even after 100 passes μ_s had risen only to 0.13 and μ_k to 0.08. Photomicrographs were made of the wear tracks at various stages of the wear process, but at no time was there evidence of excessive wear or rupture of the film. Such behavior is characteristic of a good condition of boundary lubrication.

In addition to this transfer property and low coefficient of friction, thin films of TFE on hard substrates have the ability to support high pressures. Although the yield pressure of bulk TFE is in the range of 4,000 to 6,000 psi, TFE films of the thickness described here supported pressures nearly ten times as great when the hard substrate was a metal such as brass or steel. The reason for this is that the hard substrate carried most of the load.

DISCUSSION: Frictional measurements have been reported by Shooter and Thomas[17] and more recently by Shooter and Tabor[18] using a group of plastics which had been abraded with 600A emery paper under water. Such surface treatment is believed similar to the rotary abrasion of the surfaces reported here. Their work concerned bulk TFE, polyethylene, and plasticized as well as unplasticized polyvinyl chloride. The static coefficient of friction reported by us for these three plastics and for all their sliding combinations are in good agreement with the values designated by Shooter and Tabor simply as the coefficient of friction. Since they did not report any variation of friction with sliding distance, and since it was never clear whether the values were static or kinetic, it is impossible to compare in detail our results with those they reported.

These authors chose a load range and slider radii of curvature such that the ploughing action was negligible and the real area of contact was, within a factor of two, equal to the apparent area. They showed that except for TFE the frictional force (F) between the steel slider and plastic was, within a factor of two, equal to the product of the real area of contact (a) and the bulk shear strength (s) of the plastics; i.e. (1) $F = sa$. Thus, the conclusion was reached that for plastics where (1) is applicable, shearing occurs within the bulk of the plastic and the frictional force is determined only by the bulk physical properties of the plastics. But the area of contact could be expressed by the quotient of the applied load (W) and the yield pressure (p), i.e. (2) $a = W/P$, and the frictional force is equal to the product of the coefficient of friction (μ) and the applied load, or (3) $F = \mu W$. Then it follows that (4) $\mu = (s/p)$.

We have measured the bulk shear strength of these plastics by ASTM Test Method D732-46. Yield pressures were determined by static indentation measurements in the region where the whole of the material around the area of contact was flowing plastically. Using a 3.96 mm diameter ball and loads greater than 10 kg, the yield pressure was always constant for each plastic in this load range. The values of (s) and (p) as determined from our data and s/p, are listed in Table IV, while the observed coefficients of static friction (μ_s) are given in the fifth column.

All values of μ_s listed in Table IV were obtained with rotary-abraded rather than with polished plastic surfaces. These values are preferred for comparing μ_s with s/p because: (a) the rotary-abraded surfaces nearly approximate those employed by Shooter and Tabor[18]; (b) we found the reproducibility of μ_s to be best with abraded surfaces; and (c) we found polished specimens

to have very soft surfaces so that they could not be used in comparing the shear strength of the soft surface layer with that of the bulk material.

In general, our measured values of μ_s for TFE, polyethylene, and polyvinyl chloride are in good agreement with those reported by Shooter and Tabor[18]. Since these authors did not report the molecular weights of their plastics nor their measured values of either p or s/p, much of their other data cannot be compared with ours.

The same plastic specimen was used to measure μ_s and p but another specimen was used to measure s. This may have contributed to the uncertainty in the measurement of shearing strength. The most reliable value of s is that for TFE ($\pm$ 0.8 percent), and the least reliable is for polyvinylidene chloride ($\pm$ 10 percent). Our results are in agreement with the conclusion of Shooter and Tabor[18] that, except for TFE, the measured values of μ_s are within a factor of two equal to the measured values of s/p. Hence, our work supports their conclusion that a major component of the frictional force is due to the strong adhesion which occurs between the steel ball and the plastic, resulting in a shearing in the plastic below the interface. This was clearly shown by the behavior of a steel ball sliding on polyethylene, and the associated transferring of material. Of course, as sliding proceeds, this shearing in depth is modified first by shearing between the plastic transferred to the steel ball and the bulk plastic, and later by shearing between the oriented transferred plastic and the bulk plastic.

In comparing s/p with μ_s in Table IV, we have used only the values of μ_s which most nearly represented shearing in depth of the bulk plastic, that is, the value of μ_s obtained at the beginning of sliding. With respect to TFE, the disparity between the value of μ_s and s/p makes it evident that, besides shearing in depth, another mechanism must be involved. It would appear that some slippage occurs at the metal-plastic interface, because of the low adhesion between TFE and steel. Hence the value of s/p = 0.42 corresponds to no slipping.

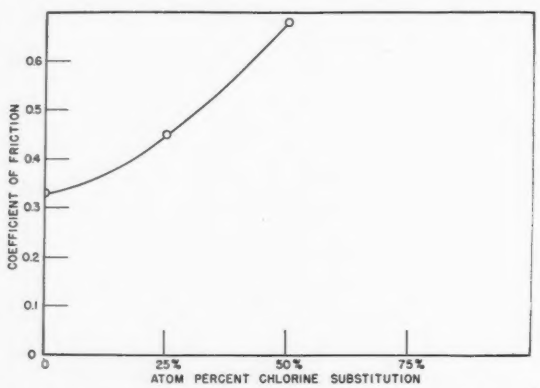


Fig. 5 Effect of chlorine substitution on static coefficient of friction of polyethylene (steel sliding on abraded plastic).

Fig. 1 demonstrates that the substitution of chlorine for fluorine atoms in polytetrafluoroethylene has a remarkably large effect on the coefficient of friction, and a rise of 4-fold in μ_s and 6-fold in μ_k occurs from the replacement of only 10 percent of the fluorine atoms. With 20 percent or more of chlorine substitution, "stick-slips" and tearing occur, and it is presumed that the adhesional force per unit area between plastic and metal has become greater than the bulk shear strength of the plastic.

Polytetrafluoroethylene, the 50-50 copolymer of tetrafluoroethylene and ethylene, and polyethylene form a series of linear polymers in which the fluorine atoms in polytetrafluoroethylene are replaced to the extent of 0, 50, and 100 percent. From the data as plotted in Fig. 4 it is apparent that the replacement of fluorine by hydrogen atoms causes an initial and rapid rise in μ_s such that the values for 50 percent substitution are practically identical with those for 100 percent. Here the replacement of fluorine by hydrogen causes effects like those of replacing fluorine by chlorine, in that each causes an increase in μ_s. However, the initial slopes of Fig. 1 are much greater than that of Fig. 4.

The fact that Fig. 4 exhibits a steep slope at 0 percent hydrogen substitution but 0 slope at 100 percent substitution deserves explanation. A rather qualitative explanation can be given. The van der Waals force field from each atom in the surface of the plastic is highly localized and nearly zero at distances of an atom diameter. The adhesional effect of the surface atoms is like the effect of inserting one strong magnet in a two dimensional array of weak magnets, all of the same orientation and strength. Obviously, a few strong magnets to every 100 weak ones will give a perceptible change in the net attractive field of this array; this explains the rapid curvature at 0 percent hydrogen substitution. When the system consists of an array of 100 strong magnets among which several weak magnets are distributed, there will result little change in the net attractive field of the array. In principle, this also explains the large slope at 0 percent chlorine in TFE in Fig. 1 and the 0 slope at 25 percent chlorine substitution in TFCE.

Polyethylene, polyvinyl chloride, and polyvinylidene chloride form a series containing 0, 25, and 50 percent of chlorine atoms substituted for the hydrogen atoms in polyethylene. Fig. 5 is a graph of μ_s for steel sliding on the abraded plastic plotted against the percent of substituted chlorine atoms. Evidently, chlorine substitution increases μ_s rapidly; and there is no evidence of the curve approaching a horizontal asymptote even at 50 percent chlorine substitution. Replacing hydrogen atoms by chlorine appears to have caused the adhesional force to exceed the bulk shear strength of the plastic so that shearing in the bulk as well as "stick-slip" appeared even with 25 percent of substituted chlorine atoms; this was true whether that substitution was made in polyethylene to form polyvinyl chloride or in polytetrafluoroethylene to form TFCE. It is remarkable that one atom of chlorine per mole of monomer should have entirely nullified the good effect of three atoms of fluorine per monomer in TFCE. Polished polyvinylidene chloride adheres so remarkably to other pieces of unplasticized material and to clean polished metals as to make it diffcult to handle and to keep free of scars and scratches. From these examples of the large rise in μ_s and μ_k caused by the addition of chlorine atoms to polyethylene or polytetrafluoroethylene, it is reasonable to predict a similar rise in the coefficient of friction resulting from the substitution of chlorine for hydrogen or fluorine in other types of polymers. The only proviso needed is that the resulting polymeric structure should permit some of the substituted chlorine atoms to be exposed at the plastic-air interface.

That steel sliding on abraded TFCE has a coefficient of friction (μ_s) which is four times greater than that of steel on TFE may be surprising. Certainly, the coefficient of friction is one of the few physical properties which is very different for these two extremely stable plastics. It is also remarkable that μ_s for polyvinyl chloride ranges from 0.45 when abraded to 0.9 when polished, while values of from 0.7 to 1.8 are encountered with polyvinylidene chloride. Evidently these unplasticized materials can abrade badly when woven into fabrics or when rubbed against clean metals as in insulators and other protective coatings. Plasticizers for these materials must obviously be chosen to improve the boundary lubricating properties. Finally, the opposite effects of chlorine and fluorine substitution in polyethylene are strikingly shown by the change in μ_s of the polished material from 0.6 to 0.9 with the addition of one atom per monomer of chlorine and from 0.6 to 0.3 with the addition of one fluorine atom.

No evidence was found of shear orientation in the material transferred to steel rubbed with TFCE, polyvinyl chloride, or polyvinylidene chloride. This does not prove that these polymers cannot be oriented by mechanical working, and there is much evidence in the literature that they can be so oriented[19]; all that can be asserted is that the adhesional force per unit area between metal and plastic is sufficiently greater than the bulk shear strength of the plastic that insufficient slipping occurs at the metal-plastic interface to cause orientation.

Such low coefficients of friction as 0.05 exhibited by thin films of polyethylene on hard substrates deserves explanation. The low friction is caused by the low bulk shear strength (s) of the organic coating and the low value of the real area of contact (a) resulting from the small deformation of the hard metal substrate. This is the same mechanism that Bowden and Tabor showed was responsible for the low coefficient of friction of a thin film of indium on steel[12]. Since indium, polyethylene, and TFE have values of s of 0.22, 0.91, and 1.62 it is not surprising that the μ_s and μ_k for films of the latter two on steel or brass should also be low.

It is worthwhile to examine quantitatively the behavior of a thin, soft film of plastic material on a hard steel substrate. Equation (1)—(4) can be applied to the sliding of a hard steel ball on a thin plastic film of bulk shear strength s_1 and mean yield pressure p_1 deposited on a hard substrate whose corresponding bulk constants are s_2 and p_2. Now, $W = p_1 a_1$, $F_1 = s_1 a_1$, and $F_1 = \mu_1 W$, where F_1 is the frictional force and a_1 the real area of contact for the sliding of the steel ball on the bulk plastic. Also, we have $W = p_2 a_2$, $F_2 = s_1' a_2$, and $F_2 = \mu_2 W$, where F_2 is the frictional force, and a_2 is the real

(Continued on page 218)

MATHEMATICAL SYMBOLS FOR LUBRICATION PROBLEMS

by J. Boyd, ASLE Education Committee*
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In the December 1951 issue of LUBRICATION ENGINEERING a proposed list of mathematical symbols for lubrication problems was published and comments regarding their selection and use were solicited. The replies which were received have been carefully studied and a revised list prepared. This list has been submitted to the Board of Directors and was accepted for use by the Society on April 15, 1953. Authors preparing papers for presentation and publication before the Society will be asked to adhere to these symbols where they apply.

Since the comments which were made were in general quite favorable, only minor changes were necessary in the preparation of the revised list. These included the following: (1) dropping the word "work" from the pervious definition of the symbol "E" and substituting the symbol "W_k" for the letter "E" as the symbol for work; (2) dropping the symbol "P_r" for power and having only the symbol "H" for this quantity.

Since it is desirable that the underlying principle involved in selecting the approved symbols be made clear and since a few comments will make the reasons for certain selections more apparent, a repetition of some of the remarks made in the previous publication is felt to be in order.

GENERAL PRINCIPLES: In setting up the symbols for lubrication the following general principles, based upon the experience of other standardizing groups, were followed:

1. It is not necessary to have a separate symbol for all the quantities which are encountered in lubrication calculations, but only for those quantities which are in common use and which would involve confusion if different authors used different symbols. It is better to keep the number of symbols as small as possible. Additional symbols can be added from time to time if the need arises.

2. Authors cannot be forced to use a given set of symbols. Their use is entirely voluntary. Thus it is desirable to permit the use of alternate symbols wherever possible provided they do not result in confusion with the symbols for other quantities.

3. The establishment of a given symbol to represent a given quantity does not mean that an author cannot use the same symbol to represent another quantity for which no symbol has been assigned. It merely means that, when dealing with quantities for which standard symbols exist, he should use the established symbol.

4. The establishment of a symbol for a quantity does not determine the units in which the quantity is to be expressed. Any consistent system of units is satisfactory.

5. Since other branches of engineering have already gone to considerable trouble to set up their own standard symbols and since the American Standards Association has done an important job of correlating the symbols for the various branches, every effort should be made to conform to the pattern already set.

COMMENTS ON INDIVIDUAL SYMBOLS: The accompanying symbols have been chosen after a careful study of various texts and papers on lubrication. In most cases there are fairly compelling reasons for the choice of symbols shown. Some cases, however, deserve additional comments.

The symbols B, b, for breadth parallel to the direction of motion and L, l, for length perpendicular to the direction of motion have to be considered together. Although, at first glance, it might seem more logical to use L, l, for length parallel to the direction of motion, most authors use B, b, for this quantity. There is good reason for this since in teaching one starts with the simple slider bearing having the dimensions B and L and then proceeds to the journal bearing having the dimensions D (diameter) and L (length perpendicular to motion). Since the meaning of D and L is evident and so well established in the case of the journal bearing, consistency makes it necessary to use L, l, for the dimension perpendicular to motion on other bearings (such as the slider bearing) also.

The letters C, c, for radial clearance may call for further discussion. Some people like to work with diametral clearances since in full bearings it is rather easily measured. However, in practically all mathematical treatments, it is much easier to deal with radial clearance, in fact in the case of partial bearings, diametral clearance does not have much meaning. The use of C/2 for radial clearance, where C is the diametral clearance, has the objection that it is too complicated for a term which must be worked with so frequently. It has also been suggested that small c be used for radial clearance and large C for diametral clearance. This is satisfactory in printed matter, but, in the classroom, it results in confusion since a large C and a small one look alike in spite of the instructor's best efforts to show a difference, and sound alike regardless of anything he can do. Since the teaching angle is very important in broadening the understanding of lubrication, it was felt that the radial clearance should be the primary unit, permitting either large C or small c to be used as a symbol.

*The Education Committee, under whose auspices the accompanying list of symbols has been prepared, comprises the following: D. D. Fuller (Chairman), A. R. Black, J. Boyd, A. F. Brewer, J. T. Burwell, Jr., W. E. Campbell, J. I. Clower, M. E. Dougherty, G. R. Dunne, W. T. Everitt, J. H. Fuller, S. Kyropoulos, C. M. Larson, R. H. G. Mathews, C. E. Pritchard, E. S. Ross, and R. Q. Sharpe. In preparing the list, the committee was fortunate to have the comments and suggestions of Mr. M. D. Hersey.

It will be noted that no symbol is included for heat, or the amount of thermal energy. Inasmuch as one is almost always much more interested in the heat rate (amount of heat per unit time), this omission is not objectionable. A further reason for leaving out a symbol for heat is that the only logical symbol for it would appear to be either h, or H which would result in conflicts with film thickness and power. The symbol here chosen for heat is q which is the A.S.A. standard for heat and thermodynamic calculations.

Because of the wide use and acceptance of $\mu N/P$ in bearing literature, it is almost mandatory that large P be reserved for the load per unit of projected area. To distinguish the load per unit area from the total load it was decided that the latter should be designated by large W.

Probably the hardest term to fit into the already accepted A.S.A standards is that for power. These standards in all the cases examined use the letter P. As already pointed out, special usage in lubrication has strongly identified P with the load per unit of projected area. It seems wise not to attempt to change this. The use of H for power, as here employed, agrees with the usage in many books and lubrication articles but of course is at variance with the A.S.A.

The alternate symbol, θ, for temperature is intended for use when time and temperature occur in the same equations. This is similar to A.S.A. usage in the Mechanics of Solid Bodies.

Although there seems to be general agreement in other fields on the use of μ for absolute viscosity, some people in lubrication prefer to use Z. For this reason, the latter term has been added as an alternate.

Mathematical Symbols for Lubrication Problems

α	acceleration, angular
g	acceleration, gravitational
a	acceleration, linear
$\alpha, \beta, \gamma \ldots$	angles
ω	angular velocity
A	area
B, b	breadth, width (par. to dir. of motion)
C, c	clearance, radial
f	coefficient of friction
α	coefficient of linear expansion
ρ	density, mass per unit vol.
D, d	diameter
e	eccentricity, brg.-jnl. center dist.
ε	eccentricity ratio, eccen./rad. clear.
η	efficiency
δ	elongation, deformation, displacement
E	energy
h	film thickness, thickness
W	force, load
F	friction force
q	heat rate, heat flow rate, heat per unit time
λ	heat transfer coefficient, heat flow rate per unit area, per degree
L, l	length (perp. to dir. of motion)
W	load
P	load per unit proj. area
G	lubrication number, $\mu N/P$, ZN/P
m	mass
J	mechanical equivalent of heat
M	moment of force
n	number
H	power
p	pressure, force per unit area
R, r	radius
N, n	revolutions per unit time
τ, s	shear stress
R	shear rate
S	Sommerfeld number, $(R/C)^2$ $(\mu N/P)$
c	specific heat
T	temperature, absolute
t, θ	temperature, ordinary
Δt	temperature difference, temp. rise
k	thermal conductivity, heat flow rate, per unit area per unit length, per degree
t	time
M	torque
u, v, U, V	velocity
μ, Z	viscosity, absolute
ν	viscosity, kinematic
V, V_L	volume
Q	volume rate, flow rate, vol. per unit time
γ	weight per unit vol.
W_k	work

Personals

J. CONWAY has been appointed an Assistant Sales Manager for Industrial Chemicals of Carbide & Carbon Chemicals Co., a Div. of Union Carbide & Carbon Corp.

Directors of the Stewart-Warner Corp. have re-elected all incumbent corporation officers and, in addition, created and filled six new Vice-Presidents as follows: J. E. Burke, W. W. Miller, D. C. Peterson, L. G. Vanderhoof, W. F. Herold, and C. J. Winkler.

The Colt-Worthington Oil Works, Inc., manufacturers of argolene lubricants, have moved their office and plant to Dickens St., Westbury, N. Y.

D. T. STAPLES has been elected President of the Tide Water Associated Oil Co.

Honan-Crane Corp. has announced the promotion of T. C. TIEARNEY to their Sales Engineering staff, servicing the company's Ford Motor Co. accounts in Detroit, Chicago, and Cleveland.

ASTM Officers for 1953-1954 have been announced as follows: President — L. C. Beard, Jr., Socony-Vacuum Oil Co., Inc.; Vice-President — C. H. Fellows, Detroit Edison Co.; Board of Directors — N. A. Fowler, General Box Co.; R. T. Kropf, Belding-Hemingway-Corticelli; T. E. Olt, Armco Steel Corp.; J. R. Townsend, Sandia Corp.; and K. B. Woods, Purdue Univ.

The Metalloid Corp. has opened a Chicago office at 753 W. 79th St., with technical field service and sales under the direction of H. SEXAUER.

C. F. HOLLAND has joined the Columbus, Ohio, office of the Honan-Crane Corp. as Sales Engineer for the central Ohio and western West Virginia territory; D. D. BROCK as Sales Engineer for the Kansas City area.

FRICTION & BOUNDARY LUBRICATION*

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Richmond, Calif.

The subject of friction and boundary lubrication is of noteworthy significance to the lubrication engineer who is responsible for providing proper lubrication for the great machine giants of our industrial world; it is of equal importance to the automotive design engineer who is ever seeking to improve vehicle dependability, luxury, and performance; and it is of paramount importance to the petroleum chemists striving to improve the lubricants for the unending multitudes of passenger cars and trucks swarming our highways. Whatever may be the diversities of our specific problems, there is one common denominator which binds us all together. That universal common denominator of all lubrication problems is the fact that in the final analysis we are all dealing with surfaces, predominantly metal surfaces, in moving contact. Even the most ardent disciples of those two patron saints of fluid lubrication concepts, Beauchamp Tower and Osborne Reynolds, must admit that at some period in the operating cycle of all moving parts we encounter the attrition of metal surfaces. It is fitting, therefore, that the nature of metallic friction and some of the phenomena of lubrication under boundary conditions be considered. On the basis of a review and analysis of the extensive published literature on this subject, the mechanism of sliding friction will be discussed as applied to metal surfaces under conditions of both dry metallic contact and boundary lubrication.

In introducing a discussion of metallic friction, it is well to recall something of the nature and contour of a metal surface. Not only is it impossible to prepare a metal specimen which has a truly flat face, such as the theoretical geometric plane, but it is exceedingly difficult to produce a surface of any appreciable size that is flat to within 100 or even 1000 times the size of an atom. In spite of the advances in machining and polishing techniques achieved by our modern technology, even the finest commercial finishes have peaks and valleys many hundreds of times greater than the range of molecular attraction of surface molecules. A metal surface defies complete description by any single parameter or method of examination, and a multiplicity of instruments and methods for studying the roughness of metal surfaces are reported in the literature[1 to 7]. Perhaps one of the most helpful means of gaining at least a qualitative appreciation of surface contour is through photomicroscoptic examination of a taper section, a technique first employed by Nelson[2]. Fig. 1 is a 10:1 taper section of a ground surface and illustrates the fact that the metal surfaces commonly dealt with in engineering practice are composed of a myriad of irregular asperities. Thus, when two such surfaces are placed in "contact," they actually touch only on the tips of a few of the highest asperities of each surface. With this as the picture of metallic contact, the importance of friction and the need for lubrication become distinctly apparent.

THE FRICTION OF UNLUBRICATED SURFACES: The earliest investigation of friction dates back to the experiments of Leonardo da Vinci about the year 1500, although his work was not published or made generally known until many years later. The first formal presentation of friction studies was by Amontons[8] in 1699 before the Royal Academy of Sciences in France in which friction was attributed to the interlocking of surface asperities on the sliding surfaces. More precisely, Amontons formulated that when two metal surfaces are in contact, the frictional force, or the force required to cause tangential motion, is independent of the area of contact and proportional to the normal force holding the two surfaces together. Thus, the coefficient of friction, being defined as the ratio of the tangential force to the normal force, is independent of both load and area of contact. Nearly a century later, Coulomb[9] in a series of careful and systematic investigations confirmed Amontons' work and added a third law of friction, namely, that the frictional force is independent of the velocity of relative motion. These three classical laws of friction have been found to be valid down through the years in most engineering applications. However, in a somewhat more theoretical consideration of friction, certain limitations and qualifications must be imposed on these laws to assure their validity. Both Coulomb and Amontons explained friction as the force required to overcome the interlocking of surface asperities. Referring to Fig. 2, which is a diagram of portions of two metal surfaces in contact, motion as indicated can occur by the lifting of the upper surface over the asperity in the lower surface. Thus, a purely mechanical explanation of friction results in which the frictional force equals W tan $\ominus$, and the coefficient of friction equals tan $\ominus$. At the present time the strongest proponent of this theory of friction is Bikermann[10 to 14], formerly of England, now of Merck and Company. In order for this mechanical explanation of friction to be valid, Mr. Bikermann limits the use of the term "friction" to a very narrow range of phenomena and excludes most of the recent observations generally conceded by other experimenters to be within the scope of friction studies. Also Strang and Lewis[15] conducted careful experiments to measure the magnitude of this lifting term, and their experimental work led them to the conclusion that only a few per cent of the total frictional work could be accounted for on this mechanical basis. Therefore, it is reasonable to look for some other factor as the primary cause of friction.

THE WELDING-SHEARING THEORY: Although much research is being done in this country on the theory of friction, probably the most extensive study of the nature of friction at the present time is that of Bowden[16] and his associates at Cambridge University in England. Professor Bowden explains the frictional force as composed of a plowing term and a shearing term. Referring again to Fig. 2, it is apparent that relative motion could take place by the asperity on the upper surface acting as a cutting tool and plowing out metal from the lower surface. Also sliding could occur through the deformation and shearing of the metallic junction shown formed over the Area A. These plowing and shearing terms have been examined in considerable detail, and Bowden and Tabor report a very interesting experiment in which they succeeded in measuring the two terms separately. They have shown that the shearing term accounts for 80% or more of the total force of friction, and consider the remainder of the force to be explained in terms of plowing. Any mechanical component due to the lifting of the asperities over one another appears to be negligible. This explanation of friction is basically supported by the work of Merchant[17], Sakmann[18 and 19], Burwell[19 to 21], Simon[22], and many others, although some workers include a "lifting" component in preference to the plowing term. The relative importance of these three terms varies to some extent with the physical properties of the specific metal surfaces in contact. For example, when hard surfaces are in contact, the plowing term will be somewhat reduced, and the lifting terms slightly increased. On the other hand, the plowing term represents a larger fraction of the total, and the lifting term disappears when soft surfaces are in contact.

Bowden's experiments also showed that the frictional force, due largely to the shearing term, is proportional to the actual area of contact. At first thought, this appears to be in contradiction to the classical laws of friction; however, this apparent discrepancy is resolved when it is realized that Bowden is referring to the area of actual contact and that the classical law of friction states that the frictional force is independent of the nominal or apparent area of contact. The difference between the apparent area of contact and the actual area of contact may be clearly understood in terms of the picture of metal surface contour already presented in which the surface is described as

*Sponsored by the ASLE Technical Committee on Lubrication Fundamentals and presented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

being covered with a random distribution of asperities and depressions. Thus, when two metal plates, for example, are in so-called "contact," they are actually touching each other only on the tips of their surface asperities. It is the aggregate area of the tips of contacting asperities that is called the area of actual contact, and this area is frequently only 1/100,000 of the apparent area of the plates. The tips of these asperities in contact experience plastic deformation as load is applied, and they flow plastically until their aggregate area is sufficient to support the load. That is to say, the area of actual contact increases in proportion to the applied load. Since the frictional force is largely the force required to shear the metallic junctions formed by the contacting asperities, it is obvious that this shearing force is proportional to the area of these junctions.

As part of his study of the friction of unlubricated metal surfaces, Bowden experimentally isolated the effects of load and area of contact to determine conclusively whether the frictional force depended on the magnitude of the load or the magnitude of the actual area of contact. By using a hard hemispherical specimen sliding over a hard metal base having varying thicknesses of soft metal overlay, he was able to vary the track width and area of actual contact while holding the load constant, and conversely, he was able to vary the load while maintaining constant area of actual contact. He was thus able to establish the fact that the frictional force is proportional to the actual area of contact and independent of load. The classical laws of friction are still valid in engineering practice because the actual area of contact and the load are proportionately and inseparably related. It is well to remember, however, that the frictional force is proportional to the load only because the area of actual contact increases in proportion to the load.

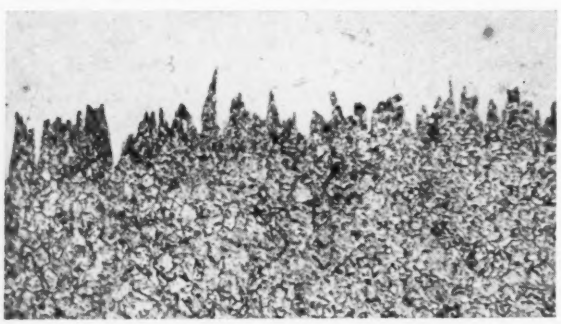

Fig. 1 Ground surface 10:1 taper section. Magnification: horizontal, 250X; vertical, 2500X.

Bowden mathematically expresses the frictional force as: $F = sA + P$ (where s = shear strength of the metal, A = the actual area of contact, and P = the plowing force). Since the shearing term is considerably greater than the plowing term, the accuracy is not seriously affected when the coefficient of friction is written:

$$\mu = \frac{sA}{W} = \frac{sA}{p_m A} = \frac{s}{p_m}$$

(where p_m is the mean yield pressure of the metal and W is the load). The coefficient of friction may be thus expressed as the ratio: The shear strength of the softer of the two materials in contact divided by the mean yield pressure of the softer material. This theory adequately accounts for the small temperature coefficient of friction since temperature affects both the shear strength and the mean yield pressure in a similar manner. It is also compatible with the observation that the coefficient of friction is approximately the same for all metals since, in general, metals with high shear strength will also have a high mean yield pressure. Thus, the most important factor in the friction of unlubricated surfaces appears to be the formation and breaking of the minute welded junctions between the tips of surface asperities.

In order to investigate the existence of welded junctions at the tips of the asperities of contacting metal surfaces, the writer placed together specially prepared specimens in which the apparent area of contact was quite localized (of the order of 1/250 sq. in.) and prepared metallographically polished sections through this region of contact. With a microscope the apparent area of contact was searched for isolated occurrences of "actual" contact between the tips of asperities. Such isolated areas of actual contact were found, and the intimacy of the bond can be observed in the photomicrograph presented in Fig. 3, the arrows indicating the original line of separation between the two specimens. A study of this photomicrograph reveals a continuous metal structure through this "bridge" from one specimen to the other. Thus, in order for sliding to occur, these bridges must be sheared. Although no attempt was made to measure the strength of these junctions, the photomicrographs taken offered vivid evidence of the existence of such bridges.

The irregular manner in which two unlubricated metal surfaces move over each other, referred to as the "stick-slip" behavior, can be accounted for on the basis of this welding theory of friction. It is well known that metal surfaces in general do not slide smoothly over each other; rather they stick momentarily, the coefficient of static friction being larger than the coefficient of kinetic friction, then slip a short distance, stick again until the moving force again exceeds the value of static friction, at which time another slip occurs, etc. Although the detail of the stick-slip behavior is markedly influenced by the dynamics of the particular friction machine being used to study the phenomena, the stick portion of the cycle is explained by the Bowden school as due to the welding or adhesion of the tip of surface asperities, and the slip occurs when the tangential force becomes sufficient to break these welds. High temperature flashes of the order of 500 to 700 C. have been observed during the slip portion of the cycle. It should also be mentioned, however, that recent work by Parker and Hatch[23] of Ferodo, Ltd., England, suggests that even during the so-called "stick" portion of the cycle, microslip is occurring. Admittedly, it is difficult to distinguish between the very small microslippage between surface asperities and simply a growth process caused by plastic flow due to the increase of tangential force.

THE ELECTROSTATIC THEORY: Another entirely different theory of friction proposed by Schnurmann and Warlow-Davies[24 to 27], of the London, Midland, and Scottish Railway, is the electrostatic theory, and it is based on the occurrence of micro-slippage during the so-called stick portion of the stick-slip cycle. According to this theory, as two metal surfaces rub together, one gains and the other loses electrons, building up clusters of electrical charges of opposite polarity on the adjoining surfaces. The surface oxides present on all metal surfaces are considered to be sufficiently poor electrical conductors to permit this charge buildup. The surfaces are then held together by the electrical attraction of unlike charges. Upon applying tangential force micro-slippage occurs, being restrained by this electrical attraction until the charge separation increases the potential sufficiently to cause an electrical discharge between the two surfaces and macro-slip results. Schnurmann and Warlow-Davies have measured those electrical discharges with a sensitive galvanometer and have found that the size of the charge depends upon the nature, pressure, and humidity of the surrounding atmosphere. In other papers Shaw[28] and also Schnurmann[29] definitely show the presence of electrostatic forces, but do not attempt to establish them as the primary cause of friction.

THE MOLECULAR ATTRACTION THEORY: Another current theory of friction is the molecular attraction theory originally proposed by Ewing[30], and supported by the extensive work of Sir William Hardy[31] and Tomlinson[32] in England. In order to understand this theory brief mention should be made of the inter-molecular forces which must be present in order for the solid state to exist. In this discussion of the solid state the use of the word "molecule" in describing the structure of a solid should be clarified. It is recognized that in the case of a metal the structure within the lattice is composed of metal ions with the valence electrons relatively free to move around through the entire solid; in a non-conducting solid the valence electrons are far less free to move, and the term "atom" should more properly be used. However, for simplicity of expression, the word molecule will be used to describe the atom in both the ionic and atomic types of solid structure.

Since the molecules of a solid are in a state of equilibrium, there must be a balance between the attractive and repulsive forces acting on each molecule. For simplicity let us consider a "solid" made up of only two molecules. These two molecules are attracted to each other by a force due largely to intra-atomic charge distribution, having the form $1/r^m$ where r is the distance between centers of the two molecules and m is an exponent having a value in the range from 3 to 6. Due to the repulsion between the inner shells of electrons in the atoms, there is also a repulsive force acting between the two molecules which has the form $-1/r^n$ where n has a value from 9 to 14. At large distances of separation between the two molecules under consideration, the resultant force, $1/r^m - 1/r^n$ (where $n > m$), is attractive, though at sufficiently large distances of separation it approaches zero. As the distance between molecular centers is decreased, this attractive force between the two molecules increases and rises to a maximum; at a distance where the simultaneously acting repulsive force becomes of significant magnitude, the resultant force begins decreasing until a distance of separation is reached where the attractive and repulsive forces between the two molecules are in balance and the resultant is zero. Closer approach of the two molecules results in a repulsive

force of great magnitude. Thus the normal spacing between molecules in the structure of a solid is the equilibrium position where the attractive and repulsive forces are in balance, thermal vibrations and other oscillatory motions being omitted as not essential to the present discussion. As a solid is subjected to externally applied load, the molecules are displaced slightly from their equilibrium positions and enter into each other's repulsion fields with the result that a large restoring force is developed which is sufficient to support the applied load.

The molecular theory of friction is based on the existence of the forces as described. When two metal surfaces slide over each other, many surface molecules of one body enter into the attraction field of the surface molecules of the other body. However, only a few molecules enter into the repulsion field of the second body. It can be shown by mathematical considerations that when two molecules come into each other's repulsion fields, that is, come into contact, and then separate, a loss of energy may occur which is manifest as friction and appears as heat. According to the theory suggested, friction is in a sense a matter of chance, since an energy loss during separation of two molescules that have entered each other's repulsion fields is dependent upon the molecules passing into an unstable region during separation. A statistical aggregate of a great number of these minute energy losses results in the frictional force. It may be remarked that on the basis of this theory the two bodies in contact become united by the formation of a continuous chain of molecules in stable equilibrium at all the points where molecular repulsion occurs. On severing the inter-molecular structure, the plucking action of one molecule on the other dissipates a quantity of energy which is independent of the velocity of relative motion. In this respect, the theory is in agreement with experiments indicating that friction is independent of the velocity of sliding. Further, only those molecules which contribute to the elastic reaction and so help for a short time to support the load pass through the irreversible or unstable state and so involve friction. This is in conformity with the observation that friction is proportional to the actual area of contact. Due to the formation of a continuous chain of molecules in stable equilibrium, it will be a matter of chance whether a molecule originally on the surface of one body remains attached to that body or is transferred to the surface of the contacting body. Again this is borne out by experimental evidence, for whenever any two metals come in contact and are separated, portions of each metal can be found on the surface of the other metal.

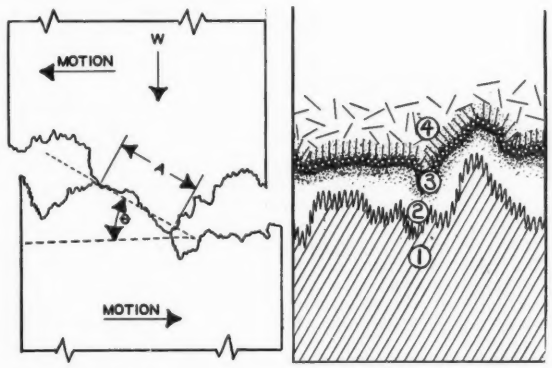

Fig. 2 (left) Profile of metal surfaces in contact.

Fig. 4 (right) The friction surface. Key: 1—Backing, 2—Bearing surface, 3—Boundary layer, 4—Fluid layer.

Although some workers in this field consider the molecular attraction theory and the welding or adhesion theory of Bowden to be in conflict, the writer fails to see any sharp disagreement between the two theories. Molecular attraction may be considered as the explanation for the adhesions or welded junctions which Bowden describes. Bowden and Tabor suggest, however, that if the molecular attraction theory is valid, any means of increasing the separation between metal surfaces in nominal contact should have the same effect. That is to say, the presence of any adsorbed gas on the surface should offer equal reduction in the frictional force. This is contrary to observation for the presence of adsorbed nitrogen or hydrogen causes little or no change in the coefficient of friction. However, the presence of oxygen materially reduces friction. They, therefore, conclude that molecular separation alone is insufficient to account for this difference; rather that surface activities and surface energies fostering or inhibiting welding are of prime importance.

The work of Sakmann, Burwell, and Irvine[19] of Massachusetts Institute of Technology on the measurement of the adhesion component in friction adds evidence to the significance of the welding-shearing theory. They found, using sliders of various hardness traversing a radioactive beryllium-copper base, that the softer the slider, the more metal it had transferred to it during sliding. Thus, the effect of increasing the area of actual contact was more significant in determining the amount of metal transfer than any possible abrading effect of hard asperities. This definitely supports the importance of the shearing term and points out the minor significance of the lifting and plowing components.

An investigation of the relationship between friction and wear made by Savage[33] and also by Whittaker[34] indicates that only a very small percentage of the work required to overcome friction can be attributed to wear. They independently conclude that the greatest portion of the frictional force is dissipated in the form of heat. Although they both worked with at least one non-metallic specimen, their conclusions can, in general, be applied to metallic wear also. Thus, a low coefficient of friction does not necessarily result in low wear. The coefficient of friction does affect wear indirectly, however, inasmuch as the coefficient of friction affects the temperature, and the temperature in turn affects the wear rate.

SUMMARY OF DRY METALLIC FRICTION: Thus, the majority of evidence presented in the literature supports the theory that the friction between non-lubricated metal surfaces is caused by the formation of minute "bridges" between the sliding surfaces, these bridges having a unit strength equal to that of the bulk metal. Whether the formation of these bridges can best be explained in terms of molecular attraction, welding, or electrostatic attraction is yet to be conclusively established.

The discussion thus far has been concerned entirely with friction under conditions of dry metallic contact. Now let us consider the action of lubricants in reducing friction and modifying some of the deleterious phenomena of dry friction.

THE FRICTION OF LUBRICATED SURFACES: The action of various substances in reducing friction may well be traced back to early primitive man smearing animal fat on rough wooden wheels and axles in order to reduce the effort required to move his crude carts. However, the earliest systematic investigation of the action of lubricants reported was by Coulomb[9] in the friction experiments already referred to in connection with dry metallic friction.

Coulomb explained the action of a lubricant in reducing friction on the basis of the lubricant filling the irregularities on the sliding surfaces and making them more "slippery." In 1910 Lord Rayleigh[35] observed the lubricating effect of a "greasy" film less than 1/200,000 of an inch thick. He was also led to believe that even thinner films would produce some "slipperiness." It was some ten years later that Irving Langmuir[36] experimentally confirmed Rayleigh's speculations by observing the lubricating effect of monomolecular layers. In 1919 Sir William B. Hardy coined the term "boundary lubrication" and began a series of classical experiments in the study of friction and boundary lubrication. Although it was not until 1922 that the SAE defined oiliness as "a term signifying differences in friction greater than can be accounted for on the basis of viscosity when comparing lubricants under identical test conditions," the first oiliness additive patent appears to have been issued in 1920 to Wells and Southcombe[37].

The work of Sir William Hardy beginning in 1919 should receive particular mention in any discussion of boundary lubrication. Hardy conducted extensive experiments studying static friction using specimens of glass, various steels, bismuth, and bronze and used approximately one hundred pure chemicals as lubricants. Many of the conclusions reached by Hardy and his coworkers stand essentially unchanged today and thus continue to pay tribute to the genius and ingenuity of this outstanding scientist. Hardy was the first to attack the Coulomb mechanical hypothesis for the theory of friction on the basis of his knowledge of surface force fields. He reasoned that according to the Coulomb hypothesis which attributed friction entirely to the lifting of one surface over the asperities in the mating surface, in the final analysis, the friction between minute asperities sliding over each other is zero. This would only be true if the equipotential surface of the edge of an asperity was a true plane, and this was known not to be true. Hardy suggested that the asperities required by Coulomb's theory actually were the atoms and molecules of the solids. Thus, the frictional force is explained as the force required to move normal to the equipotential plane in the force field. Hardy considered a lubricant to modify the force field, and in turn he believed the lubricant configuration to be influenced and determined by the force field. Hardy then postulated that slip may occur by either of two mechanisms: (1) as a result of the general breakdown of lubricant configuration and orientation, or (2) as a result of slippage between lubricant layers, the general lubricant structure

remaining essentially unchanged. Although Hardy's work may not be considered as all-inclusive or as giving a complete picture of boundary lubrication in the light of present knowledge, still Hardy's conclusions strike very near to the heart of present boundary lubrication theory.

GENERAL BOUNDARY LUBRICATION CONSIDERATION: Discussion of boundary lubrication is sufficiently complex to make a simple outline of the subject extremely diffcult, and it may be well to consider first some general facts regarding lubricants and boundary conditions. It is, of course, essential that we understand exactly what is meant by the term "boundary lubrication." Whereas the term "dry metallic friction' refers to the complete absence of a lubricant and the term "full fluid lubrication" refers to a condition where the lubricant is present in sufficient quantity to separate completely the sliding metal surfaces, the terms "boundary lubrication" and "thin-film lubrication" refer to conditions where lubricant is present, but due to the loading and geometry of the mating surfaces and the quantity of lubricant, complete separation of the sliding surfaces is not possible. In thin-film lubrication a portion of the load is carried by a fluid film, and a portion is carried by metal-to-metal contact; while in boundary lubrication, which may be considered an extreme case of thin-film lubrication, fluid effects are negligible, and the entire load is borne by an extremely thin boundary layer of lubricant. Thus, no hydrodynamic effects are present under boundary lubrication conditions.

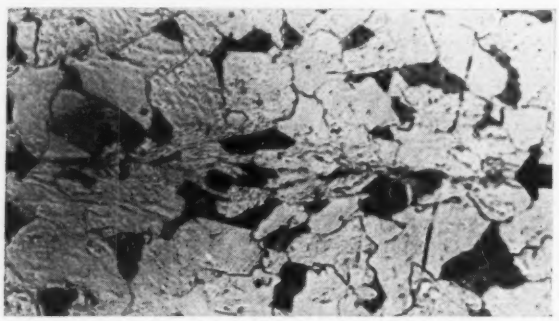

Fig. 3 Photomicrograph of the welded junctions formed between contacting metal surfaces. Magnification: 400 X.

The roughness of metal surfaces becomes of particular significance under boundary conditions as reported by Burwell[38] at MIT. The coefficient of friction generally increases with increasing surface roughness, the effect being somewhat more pronounced for pure mineral oils than for fatty acids. An interesting observation by Finch[39] of the Imperial College of Science and Technology, London, relates to the severity of wear when relative motion between mating surfaces is parallel to the lay of the surface finish as compared to motion perpendicular to the lay. At first thought, it seems surprising that the heaviest wear is observed with motion parallel to the lay. However, this may be logically explained by the fact that when sliding parallel to the surface ridges, an asperity experiencing actual metal-to-metal contact has, in general, much farther to travel under this severe condition before contact can be severed by passing into an adjoining "valley" in the surface. This causes higher surface temperatures, more deformation and plastic flow, more metal transfer, and greater wear.

Throughout the years of extensive experimentation on lubricants with literally hundreds of different chemicals tried as lubricants, there still has never been found any universal lubricant. That is to say, lubricants are specific in their action, and the behavior of a lubricant must always be evaluated in connection with the nature of the sliding surfaces for which lubrication is sought. It is also universally accepted that metallic contact between sliding surfaces is inevitable in spite of applying the fine lubricants which are now available. Through proper lubrication the amount of metal-to-metal contact and subsequent metal transfer may be reduced to a very small fraction of that experienced in the absence of lubrication; however, the complete abolition of metal transfer and wear remains an insurmountable limitation. Gregory[40], Moore[41], and Rabinowicz[42], all of Bowden's laboratory in Cambridge, England, as well as Sakmann, Burwell, and Irvine[19] and others in this country, have all positively demonstrated the existence of metal transfer through the use of radioactive tracers. Additional evidence supporting the occurrence of metal-to-metal contact even in the presence of a lubricant is obtained from the observations of Schnurmann[43], Bowden[44], and others when a sensitive galvanometer is connected between the sliding friction specimens. As the surfaces slide over each other sharp deflections of the galvanometer are noted. If a glass plate is rubbed with a metal slider and the region of contact continuously observed by looking through the glass plate, distinct luminous flashes are observed. Schnurmann attributes these flashes to electrostatic discharges and considers them to be substantiating evidence supporting his electrostatic theory of friction; while Bowden and others of his group explain their observations as high temperature flashes occurring during the shearing of isolated welded junctions.

THE FRICTION SURFACE: In order to facilitate the discussion of boundary lubrication, it will be helpful to refer to Fig. 4, a diagram showing the essential four elements of a friction surface: a metal backing, a bearing surface, a boundary layer, and although not to be included in our present discussion, a fluid layer. Each of these elements will now be considered in somewhat further detail.

(A) THE BACKING: The purpose of the backing is to determine the geometry of the particular part or specimen under consideration. It must have sufficient strength to support the load to be placed upon it without serious deformation, and it must have other specific mechanical and physical properties as required by the application. The coefficient of friction of unlubricated metal surfaces is always high, and under such conditions, wear, metal transfer, and other surface damage are extremely severe. Thus, it is readily apparent that such surfaces would be totally unsatisfactory as sliding surfaces in our modern machines.

(B) THE BEARING SURFACE: The purpose of the bearing surface is to provide a friction surface having more desirable properties which will result in a lower coefficient of friction and a lower wear rate than clean backing metal. This may be accomplished in more than one way. A secondary material may be purposely applied to the backing, for example, as in a babbitt bearing, or the face of the backing itself may be converted into a more suitable bearing surface. The nature of the bearing surface, may, in part, be determined by the surface finishing to which the specimen has been subjected. For example, Vernon, Wormwell, and Nurse[45] of the Department of Scientific and Industrial Research in England have shown that polishing 18/8 stainless steel enriches the chromium content of the surface. This has been confirmed by other experimenters and extended to include other alloys, and, in general, it is found that polishing a metal alloy enriches the proportion of one or more of the constituents in the polished Beilby layer. In order to provide the maximum benefit, the bearing surface should be somewhat softer and of lower melting point than the underlying backing; that is to say, a surface layer of low shear strength must be present on the backing.

The thickness of this fusion layer is an important factor in determining its effectiveness. If it is too thick, the effect of the underlying backing becomes negligible, and upon contacting a mating surface plastic flow and surface deformation occur resulting in a large frictional force in spite of the material being of low unit shear strength. If the layer is too thin, it fails to afford sufficient protection to the backing and the frictional behaviour approaches that of the backing alone. Ideally, the thickness of the fusion layer should be just that of the maximum tolerable wear, so that in addition to achieving a surface of low shear strength and satisfactory durability, the strong backing supports the load and prevents gross deformation and excessive increase in the area of contact. In terms of the previously stated formula for coefficient of friction, s is minimized and the increase in A is controlled within tolerable limits, so that the product sA is significantly smaller than it is for either the backing alone or for the bulk soft material.

The bearing surface is frequently composed of a metallic oxide, and the properties of the specific oxide determine the quality of the resulting surface. If the hardness of the oxide layer is approximately equal to the hardness of the backing material, a good bearing surface results. An example of this would be copper oxide (CuO) which has a Mohs hardness of 2.5 to 3.0 and metallic copper, having a hardness in the same range. However, if the hardness of the oxide layer is considerably in excess of that of the backing material, the oxide fractures easily and provides a very poor bearing surface. This is true of aluminum oxide, having a Mohs hardness of 9, compared to metallic aluminum with a Mohs hardness of 3.5. The structure of oxide layers being isotropic, having equal strength in all directions, and exhibiting good adhesion to the backing results in oxides being reasonably good bearing surfaces.

The melting point of the bearing surface is of considerable importance. It should be less than the melting point of the backing so that with the generation of frictional heat, the bearing surface can soften and flow slightly at a temperature well below the softening point of the backing. It is in this regard that oxides fail as bearing surfaces and the advantages of sulfides, chlorides, and phosphides are evident. To cite simply one

example, iron has a melting point of 1535 C., Fe_3O_4 1538 C., and Fe_2O_3 as high as 1565 C., whereas FeS melts at 1193 C. Thus through a sulfide pretreatment of machine parts, or active sulfur in the lubricant chemically attacking the surface, a highly desirable fusion layer of metallic sulfide is formed on the surface and good frictional properties result. A boundary lubricant may be considered as simply a carrier of contaminant, whether it be oxygen, sulfur, or phosphorus, to the high temperature zone of contact. Finch[39] attaches particular significance to the oxide layer and considers the primary function of the oil to be that of carrying oxidant to the region of contact. Were it not for the oxygen, traces of sulfur dioxide and other contaminants found in normal atmosphere, the problems of the lubrication engineer would be manyfold more difficult.

(C) THE BOUNDARY LAYER: In the boundary layer an attempt is made to reduce further the shear strength of the surface layers through the addition of a lubricant. Although varying considerably in relative importance depending upon the particular service conditions, there are two fundamental requirements for a good boundary lubricant: (1) a chemically bound film of low shear strength must be formed and (2) a physically adsorbed film must be present. The chemically bound film primarily reduces wear, while the reduction in friction is largely due to the physically adsorbed film. These two functions may be accomplished by a single compound or accomplished separately by two components in a boundary lubricant. Both of these factors are important, and neither one appears sufficient in itself to eliminate the need for the other.

The chemically bound film may be a chloride or a sulfide formed by the chemical action of active chlorine or sulfur in the lubricant, and their function has already been discussed in connection with the bearing surface. Another effective group of chemicals for use as boundary lubricants is the fatty acids. These compounds have been studied extensively by many experimenters, and it is generally agreed that the fatty acid itself does not provide the observed beneficial behavior except as it reacts with the metal to form a soap. This is perhaps most strikingly illustrated by the behavior of a fatty acid as the ambient temperature of the sliding system is increased. There is a distinct transition temperature above which there is a significant decrease in the lubricating efficiency of a fatty acid. This transition temperature is above the melting point of the fatty acid and corresponds to the melting point of the metallic soap. Gregory[46] has shown that the behavior of a fatty acid depends upon the activity of the metal surface, and if chemical action between the fatty acid and the metal does not occur, the fatty acid is no more effective as a lubricant than a normal paraffin.

Some interesting work by Tingle[47] at Cambridge provides additional information on the action of fatty acids, particularly the significant influence of water in determining the behavior of the acids. Conducting his experiments in a carefully dehydrated atmosphere, Tingle used a 1% solution of lauric acid in paraffin oil as a lubricant and found the coefficient of friction to be essentially the same as for pure paraffin oil, the presence of the 1% of fatty acid being ineffectual in reducing the friction. Even if he allowed the 1% acid solution to remain in contact with the metal many hours at a temperature of 100 C., still the lauric acid was ineffective. Further, he prepared a thick film of oxide on the metal and again attempted lubrication in a dehydrated atmosphere with a 1% solution of lauric acid in paraffin oil, and again he found the fatty acid to have negligible effect. However, by introducing a slight amount of moisture into the surrounding atmosphere, the commonly observed beneficial effect of lauric acid was readily apparent. He concluded that water was essential to the chemical attack of the metal by the fatty acid to form a soap. He suggested two possible explanations to account for the importance of water. First, the water may assist in the ionization of the fatty acid, or, what he believed more likely, the water reacts with the oxide to form a loose hydrated oxide or a hydroxide which in turn is acted upon by the fatty acid.

Campbell and Thurber[48], working at the Bell Telephone Laboratories, found an apparently contradictory condition to prevail inasmuch as they observed, using normal paraffins and even a fatty acid as lubricants on surfaces of steel, brass, and glass, that water caused an increase in friction compared to that measured in dry air. This apparent contradiction may be resolved by carefully reviewing the data and experimental techniques presented in the two papers. Campbell and Thurber found that the presence of moisture produced the greatest increase in friction when chemical action between the lubricant and specimen was impossible—for example, when the specimens were glass or when the lubricant was a straight chain normal hydrocarbon—and this is quite compatible with Tingle's findings. Under these non-reactive conditions lubrication is due exclusively to a physically adsorbed film, and since the normal hydrocarbons are non-polar, the highly polar water molecules are preferentially physically adsorbed on the surface of the metal, interrupting the hydrocarbon film and thus reducing its lubricating efficiency. Campbell and Thurbers' observation of a small increase in friction when comparing moist and dry conditions using oleic acid on steel is probably due to their method of preparing a "dry" surface. Their "dry" surfaces appear to be comparable to Tingle's "wet" ones, and their "moist" surfaces had an excess of water present. Tingle started with a surface strictly free of moisture, added a small amount of water to the surrounding atmosphere, and found the lubricating efficiency of a fatty acid to be improved. Campbell and Thurber, in spite of their drying procedure, started with a surface having a slight trace of moisture present, added excessive moisture, and observed a rise in the coefficient of friction. Rather than being contradictory, the two papers are quite compatible.

The effect of water in modifying the action of a fatty acid on steel surfaces clearly illustrates the importance of the two previously stated requirements of a good boundary lubricant. When water is totally excluded, the fatty acid is unable to react with the metal surface to form a chemically bound soap film, and the friction is relatively high in spite of the formation of a good physically adsorbed film. A small amount of water permits a chemical reaction to proceed at the metal surface with the formation of a soap film which together with the physically adsorbed film provides good lubrication. If the amount of water present is further increased, the preferential physical adsorption of the water interrupts the physically adsorbed lubricant film so that despite a satisfactory chemically bound soap film the friction is again increased. Thus, under normal atmospheric conditions fatty acids are quite effective in lubricating copper, steel, and other active surfaces; however, they are no more advantageous on platinum, silver, and other non-reactive surfaces than normal paraffins.

When chemical reaction with the metal surface is not possible, either due to the nobility of the metal or inactivity of the lubricant, lubricating efficiency is determined by the strength of the physically adsorbed film. X-ray diffraction and electron diffraction studies have contributed much to our knowledge of physically adsorbed films. Normal paraffins generally form ortho-rhombic structures with the hydrocarbon chains normal to the mental surface. It is further observed that the structure remains the same throughout the entire depth of the film. Fatty acids, however, being polar molecules, form a somewhat different film structure. The first mono-layer is oriented perpendicular to the metal surface. However, subsequent layers generally crystallize in their normal mono-clinic form, with the carbon chains steeply inclined to the surface. Although molecular polarity is important in determining the strength of adhesion of a lubricant to a metal, the cohesive forces between adjoining hydrocarbon chains are also important in determining lubricating efficiency. This is evidenced by the fact that paraffins, esters, and alcohols lose much of their effectiveness as boundary lubricants above their bulk melting points. Solid paraffins, though only weakly adsorbed on a metal surface, are better lubricants than liquid alcohols in spite of the fact that the highly polar alcohol is strongly adsorbed on the surface. Even fatty acids, which form well-oriented physically adsorbed films, are not good lubricants on non-reactive surfaces above their bulk melting points. Thus, the cohesion of adjoining molecules in the lubricant is at least as important as the strength of their adsorption on the surface. The importance of cohesive factors and the necessity for lubricant molecules to form a close-packed layer is again evidenced by the fact that straight-chain hydrocarbons are generally better boundary lubricants than aromatic ring hydrocarbons.

It is not the purpose of this paper to discuss in detail the efficiency of specific lubricants, but some indication of the magnitude of the coefficient of friction under various conditions is in order. The highest friction is observed with unlubricated metals outgassed in a high vacuum, and under such conditions the coefficient of friction ranges from 5 to 100 and even on to complete seizure. If inert gases are added to the system, no change is noted; however, if oxygen is present but the surfaces otherwise unlubricated, the friction coefficient drops to the order of unity. The use of a mineral oil as a lubricant lowers the friction to approximately 0.2, and a long chain fatty acid, on reactive surfaces, further decreases the coefficient of friction to 0.1 or less. In general, kinetic friction is lower than static friction, and at high sliding velocities a further small reduction may be noted. Although considerable variation may be found in the values of the friction coefficient depending upon the nature of the surfaces and the specific lubricant, the values given at least serve to indicate the order of magnitude of friction under various conditions of boundary lubrication.

It was originally observed by Hardy and has been verified by many others since Hardy's work that the lubricating efficien-

cy of hydrocarbons increases with increasing chain length for any given series of hydrocarbons, causing a reduction in the surface damage as well as the coefficient of friction.

Beeck, Givens, and Smith[49] explain an observed decrease in the coefficient of friction as the sliding velocity increases beyond a certain critical value as due to the oriented polar molecules inducing a wedging action and promoting the formation of a fluid lubricant film. On the basis of these and other observations they have formulated an interesting quasihydrodynamic theory of friction; however, this theory more properly relates to the domain of thin-film lubrication rather than to the present discussion of boundary lubrication. The importance of this effect, however, is quite generally recognized, and in all experimental investigations of pure boundary lubrication care must be exercised to prevent the formation of a fluid film and subsequent hydrodynamic effects from interfering with the results. In an attempt to exclude hydrodynamic factors, Burwell[50] has suggested the use of a quantity which he terms an "incremental friction coefficient" derived from the friction versus velocity curve. Although not as yet widely used or accepted, this factor may at least be helpful to apply as a "test" to identify and eliminate discrepancies in observed data caused by hydrodynamic effects.

THE EFFECT OF SURROUNDING ATMOSPHERE: Gwathmey, Leidheiser, and Smith[51] working at the University of Virginia have reported a series of very interesting experiments investigating the influence of surrounding atmosphere and the effect of crystal plane on corrosion, rate of oxide formation, wetting, friction, wear, and other processes important in lubrication. Working with large, single crystals of naked metal surfaces in an inert atmosphere, they found extremely high wear as would be expected. The addition of 0.2% oxygen in a nitrogen atmosphere reduced wear 900 times. Using similar surfaces and mineral oil as a lubricant, the wear rate of a copper sphere through a lead film was 200 times faster in hydrogen than in air. Bowden and Young[52] in a recent paper also discuss the effect on boundary lubrication of the extremely thin surface films formed from atmospheric gases. These experiments and other of a similar nature serve to emphasize the importance of the contaminating gases normally found in the atmosphere and the effect they have in determining lubricant behavior.

The literature on the subject of friction and boundary lubrication is much too voluminous to permit mention of the contribution of each investigator studying in this field, and it would even be impossible to include every paper in an extended list of reference; however, a few additional interesting papers not already specifically mentioned are included in the bibliography. [53 to 77]

SUMMARY: A survey of the extensive literature on the subject of friction and boundary lubrication has revealed a general agreement that the primary mechanism of sliding friction is the formation and rupture of minute "bridges" between the asperities of the sliding surfaces. Whether the formation of such bridges can best be explained in terms of welding, electrostatic attraction, or molecular attraction is not definitely established. The function of a boundary lubricant is to reduce the strength and number of these bridges. In terms of the welding theory of friction, the action of a boundary lubricant is twofold. The lubricant must be chemically adsorbed onto the metal to form a thin fusion layer having a lower shear strength and lower melting point than the underlying metal, and the lubricant must also form a physically adsorbed layer of closely packed, oriented molecules. Sliding thus occurs largely between the physically adsorbed monolayers on the mating surfaces which offer extremely low resistance to motion. At such points where the physically adsorbed monolayer is ruptured, the chemically adsorbed film inhibits the formation of metal-to-metal junctions with consequent reduction in friction and wear.

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(LUBE LINES, Continued from page 189)

tion of Illinois Tech.; C. E. Schmitz, Crane Packing Co., will be Associate Director. J. G. Duba, Illinois Tech., and Conference Secretary, has announced the following faculty members as Committee Members: F. W. Edwards, R. E. Peck, and V. L.

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(BOWERS, CLINTON, ZISMAN, Cont'd. from page 208)

area of contact for the ball sliding on a thin film of plastic deposited on a steel plate, and s_1' is the shear strength of the thin film of plastic under compression between the ball and steel plate. Therefore (5), $\mu_2/\mu_1 = (s_1'/s_1)\ a_2/a_1 = (s_1'/s_1)\ p_1/p_2$. Equation (5) should be as correct (within a factor of two) as the Shooter and Tabor Eqs. (1) — (4) on which it is based.

The ratio s_1'/s_1 in Eq. (5) can be considered equal to 1 for all practical purposes unless extremely large unit loads are used. Since $p_2 >> p_1$ for soft plastics on hard steel, then $a_2 << a_1$; hence, it follows from Eq. (2) that $\mu_2 << \mu_1$. In Table V are given the calculated and observed values of μ_s for a steel ball sliding on a thin plastic coating applied to a hard steel plate. The calculated value of μ_s was obtained by substituting in Eq. (5) the yield pressure of the steel plate, the yield pressure of the plastic, and the static coefficient of friction of the steel ball on the bulk plastic as given in column 5 of Table IV. The mean yield pressure of the steel plate used was 40,000 psi (29.1kg/mm^2). Except for TFE, the calculated static coefficients of friction are in reasonably good agreement with the observed values. It is interesting to calculate μ_s for TFE from Eq. (5) by using the theoretical value of $\mu_s = s/p = 0.42$ obtained from the fourth column of Table IV. The resulting calculated value of $\mu_s = 0.06$ is in good agreement with the observed value of 0.04 for a thin film on steel. We pointed out earlier that μ_s and μ_k had the same value of 0.04 for a film of TFE on steel and indicated that this was due to the instantaneous transfer of TFE on contact with the steel ball. As it seemed possible that this transfer property made it more desirable to use the kinetic rather than the static values of the coefficient of friction in Eq. (5), we have computed μ_k accordingly from the value of $\mu_k = 0.04$ of steel on bulk TFE. But the resulting kinetic coefficient of 0.004 is only one-tenth of the observed value. Therefore, it is concluded that the coefficient of friction can be computed reasonably accurately from Eq. (5) for each of the plastics studied except TFE. The low values calculated for TFE are believed to be good evidence that considerable slipping at the interface between the TFE film and the sliding steel ball takes place.

In utilizing such thin films as dry lubricants, it is obviously important that they should not be readily ploughed through or scraped off the metallic slider so as to cause metal-to-metal contact. It is here that the transfer property of each of these plastics becomes especially valuable, because wherever a small hole develops in the film, the movement of the slider should tend to cover it with transferred polymer. In short, useful lubricating films will, under ordinary loads, tend to be self-maintaining or self-healing. This certainly will not be true of TFCE, polyvinyl chloride, or polyvinylidene chloride. Hence, of the plastic materials studied thus far, this ease of material transfer combined with low shear strength is confined to polyethylene and TFE. The additional properties of low adhesion and remarkable heat stability make TFE more suitable for many applications of dry lubricants. When TFE is used in a system that permits the ultilization of its transfer properties, the condition is essentially that of TFE sliding on TFE so that the coefficient of friction is 0.04. This is considerably less than μ for such familiar dry lubricants as molybdenum disulphide and graphite, which have values varying from about 0.09 to 0.12. These two materials used as additives in TFE cannot be expected to cause a further reduction in friction. However, they may find some application when so used at temperatures above 700 F. where the TFE films decompose.

It is obviously important in applying TFE films as dry

lubricants to form a secure bond between the film and the metal. Our experimental results have shown that thin films of TFE prepared with a commercial suspensoid in water can be successfully bonded to steel, brass, copper, bronze, nickel, monel, chromium, and (18 — 8) stainless steel. Many other metals that generally are difficult to lubricate because of galling have been coated also with TFE films. These include titanium, magnesium, aluminum and some of its alloys. A good lubricating and protective coating can be applied to aluminum without anodizing. However, care should be taken when preparing films on aluminum alloys containing zinc, because a chemical reaction occurs with some suspensoids.

TFE films of 0.1 to 0.05 mils thick were found to be continuous and selfhealing under the conditions of friction and wear reported here. But it should be remembered that to obtain the most durable lubricating films, the applied pressure should not exceed the yield pressure of the bulk plastic (4000 - 6000 psi). Where the useful life involves a more limited number of sliding cycles, much greater unit loads can be applied. We have found it possible in some equipment to employ loads of 40,000 psi. Uniform thin films on the hardest possible backing should be used to obtain the lowest coefficient of friction. Where rust prevention is not an important consideration, films as thin as 0.00005" give satisfactory results as boundary lubricants.

There are many applications of TFE coatings to the lubrication ef equipment such as the slowly sliding or rotating parts in hydraulic and pneumatic actuators, valves, and other components of aircraft, ship and submarine control systems, in guided missiles, ordnance equipment, instruments, computing machines, typewriters, radios, and radars. TFE films should find some applications at higher sliding velocities, but the life will be increasingly limited as the product of the velocity and the load increases; at high speeds the high temperature generated may cause rapid plastic flow and deterioration of the protective coating. TFE has proven useful at temperatures ranging from —100 F. to 500 F. and undoubtedly operation could be obtained for a more limited time at much higher temperatures. However, operation above 700 F. would be very limited by the thermal decomposition of the TFE.

In lubrication practice it is sometimes desirable to confine a liquid to a specific area. The use of smooth TFE films in such cases should be helpful in preventing the creeping and the spreading of hydraulic fluids and lubricating oils. These films on the metal parts of hydraulic and pneumatic systems are useful also in lubricating "O"-rings as well as preventing their adhesion to the seat which is so commonly the cause of packing failure.

TFE coatings on iron, steel, copper, brass, bronze, aluminum and magnesium also make excellent corrosion preventives and preservatives because of their chemical inertness and their remarkable organophobic and hydrophobic properties [7, 8, 9, 10]. For maximum protection it is desirable to bake a second film onto the first to cover pinholes. With two coatings totaling 0.4 mils thick we have been able to protect SAE 1020 steel against a spray of 20 percent salt water at 100 F. for 100 hours without any evidence of corrosion. Completely exposed steel specimens on the roof of the laboratory for six months during winter and spring were unaffected. Thin coatings of TFE are useful also in preventing electrical contact of dissimilar metals; this is a valuable method of combating electrolytic corrosion in hydraulic systems, water tanks, and etc.

Wright[20] has recently shown that when steel rubs against bulk TFE no "fretting wear" or "fretting corrosion" occurs; it is entirely possible from the friction and wear results reported here that thin films of TFE bonded to metals can be used successfully for preventing fretting damage.

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(RYDER, Continued from page 203)

proposed which gives a graphic measure of the oil qualities which seem most important in turbine engine operation. The adoption of some system of rating will take oil out of the mouse milk category and put oil selection on a more logical basis.

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COMMENTARY by D. W. Bedell, Esso Development Co.: I wish to emphasize the need for synthetic oils in turbo-jet and turbo-prop engines since there are certain elements in the aviation and allied industries who apparently do not recognize that a product of this type is necessary to successful engine operation. The power output per engine has increased tremendously during the last 25 years, the largest increase in power occurring in the switch from reciprocating engines to turbo-jet engines. With the continued increase in power output of the turbo-jet engines now in production, and under development, we have had increased requirements so far as the high temperature which the oil must withstand. Bearings must be lubricated which are located in the vicinity of the burners and temperatures in these bearings may be in excess of 450 F. If the operating temperature range for which such oils as Grade 1010 are suitable is compared with the range for the synthetic oil, it will be noted that the lubricant supplier has been able to gain both on the low temperature and high temperature ends by going to the synthetic product.

Of great concern is the volatility of the lubricant, especially in long range bomber operation. It has already been pointed out that engines now operating on Grade 1010 can gain economy by switching to the synthetic oil since the consumption is roughly 24 to 1. The vapor pressure of the presently approved MIL-L-7808A lubricant at 300 F., is 0.1 mm. of mercury whereas the Grade 1010 has a vapor pressure of 25.0 mm. of mercury at the same temperature. Naturally, a decrease in consumption means a decrease in weight of oil required per flight and a corresponding increase in payload.

Thermal stability is another factor which must be evaluated for turbo-jet and turbo-prop operation. There are several ways of doing this and Pratt & Whitney has chosen the panel coking test described in Mr. Ryder's paper. It is believed that this is a satisfactory quick test which may also be applicable for lubricant manufacturing control purposes. However, it would seem advisable to define the "reproducibility" of this bench test. On the basis of a limited amount of laboratory tests which we have conducted, it would appear that small differences in coking tendency of oils cannot be determined by this equipment as it now exists, but the difference between an oil giving 100 mg. of coke and one giving 400 mg. or 2000 mg. of coke can be detected. If the use of this bench test is to be extended, a co-operative program of some sort should be set up to standardize the units and define the reproducibility more closely. Such things as the measurement of splash rate could be standardized and simple techniques developed for checking these units.

It is realized that a good deal of thought and work went into the development of the Ryder Gear and Lubricant Tester and the result of this effort is a unit which appears to be a very satisfactory research tool for lubricant evaluation. For the purposes of refinery control, we would like to see a simpler and cheaper test developed which would correlate with the Ryder Gear and Lubricant Tester. It would also seem advisable to establish a reference oil for use with the Ryder Gear and Lubricant Tester so that minor variations in test gears could be eliminated as a factor in test results, and ratings could be expressed as a percentage of reference oil. This system has worked out satisfactorily in England where the I.A.E. Tester is used for evaluation of the load-carrying ability of synthetic oils. Again, a co-operative program to standardize the test units would seem advisable if the use of these units is to be extended.

Concerning the rating system discussed in Mr. Ryder's paper, we would be very glad to see a system of the type discussed tried out. However, as long as specification buying is continued as at present, the majority of purchasers will buy the cheapest product qualified under the specification rather than pay a premium for additional quality. It would seem preferable to have the specifications changed at regular intervals to eliminate the less desirable products as developments progress.

We do not believe that the panel coking machine as it now exists would be satisfactory for establishing the high temperature limit for synthetic oils. All oils which qualify under MIL-L-7808A would have ratings in excess of 600 F., by this proposed procedure. There is probably not more than 150 F., difference between the top visualized limit for organic compounds as lubricants and this 600 F., figure. It would certainly be diffcult to talk a purchaser into paying a premium for an additional 10 F., on the high temperature limit although this may have cost the lubricant supplier a considerable amount in materials and development costs.

COMMENTARY by C. R. Johnson, Shell Oil Co.: Literally hundreds of bench tests have been developed throughout the industry to evaluate various performance characteristics of lubricants. Many of these bench tests are retained in specifications long past their useful life due to lethargy or pride of authorship. The now discredited steam emulsion test is a good example of a method used past its retirement age. It is most important that test methods be a subject of continuing review through co-operative efforts of the engine manufacturers and lubricant suppliers to assure the rapid adoption of improvements in engine design and improved lubricants.

Ideally, a reduced scale test must either involve only a change in scale without change in any significant conditions or it must involve only changes in conditions to which the response is known for the materials to be tested. An example of the

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former is the small-scale gear testing unit which rates load-carrying capacity in terms of unit loading as does a full-scale unit. The latter category is exemplified by various oxidation and corrosion tests which are found to correlate with full-scale performance. In this case, however, favorable experience in rating one group of materials such as mineral oils cannot be assumed as applying to another class of materials such as additive or synthetic oils.

Regarding the details of several specific rating methods discussed by the author:

(1) The four ball apparatus is essentially a research tool designed to provide a means of studying boundary or extreme pressure lubrication with a minimum of complicating mechanical variables including hydrodynamic forces encountered in most applications. It is practically insensitive to viscosity in the EP range and due to its slow rubbing speed and high loads would not be expected to represent many practical cases of lubrication. Recognizing these limitations the research worker still finds this machine an extremely useful tool in the study of various aspects of boundary lubrication such as wear, friction, seizure and weld prevention.

Various agencies have selected different operating conditions and different means of expressing the results of the four ball apparatus. The flash temperature parameter concept applied to this work appears to be a useful tool although it cannot be expected to provide correlation with all gear rig results which in many cases do not themselves correlate.

(2) The coking test identified certain high temperature characteristics of lubricants, but in considering a wide variety of lubricants, the nature of the residue is important. Some materials form hard deposits which interfere with lubrication, whereas others form soft plastic materials capable of lubricating a bearing during short periods of loss of oil flow. There is opportunity here for further development of test methods capable of characterizing more completely the performance of lubricants.

(3) Surely a simple method is needed to identify the fields of application of lubricants. The proposal to define low temperature performance by the temperature at which a certain viscosity is reached appears sound since viscosity has been shown to be the predominant criterion.

Identification of the high temperature limit is much more complex. Stability, corrosion, volatility are only some of the factors to be considered. At this time there is no test which will evaluate all of these important variables. The author's suggestions should certainly stimulate work in this direction.

COMMENTARY by J. P. Simon, General Electric Co.: There are three comments we wish to make. First, more attention should be given to the compatibility of synthetic fluids with non-metallic as well as with metallic materials. Present specifications contain a metal corrosion test, but little attention is given to plastics, rubber, paint, etc., with the consequence that present fluids seriously attack even the most resistant of these materials. Making the oil less critical in this respect would increase its acceptability tremendously.

Secondly, while cost as such is not a serious problem with jet engine lubricants, high cost usually reflects manufacturing difficulty, limited raw materials, and the like. Availability of both raw material and of processed oil in time of war is an essential quality of jet engine lubricants and should not be overlooked.

The third and most important comment, concerns the four-square gear and lubricant tester. We agree that present laboratory test apparatus is unable to measure the ability of a lubricant to perform its prime function. Laboratory data are seldom able to be correlated with field experience. The four-ball wear test is probably most indicative of this. At times the trend of qualitative data received from this apparatus is reversed in the field. A simple, effective bench test has long been sought by our laboratories, and the four-square gear and lubricant tester is a long step in this direction.

However, correlation of the data from this tester with our experience has not been good. Note the following data: (1) With ANO9 grade 1010 oil, one laboratory produces scoring on this apparatus with an applied load of 120 pounds per inch of face width or 43,000 psi Hertz stress. On a similar though more elaborate apparatus in our Engineering Division, over 3200 pounds per inch, or 225,000 psi Hertz stress can be carried by this oil, without scoring. (2) With synthetic lubricant MIL-L-7808, the same laboratory produces scoring at 1885 pounds per inch, or 172,000 psi Hertz, whereas our work indicates the oil can carry the full 3200 pounds per inch. Equally impressive is that the four-square test was conducted at an oil temperature of 165 F., whereas our test was conducted at over 300 F.

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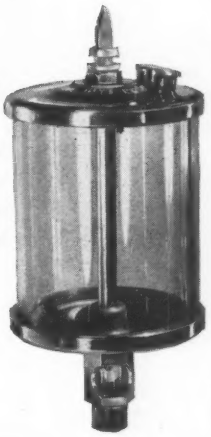

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mechanical limitation of the apparatus. We have noted the following: (1) The apparatus is too rigid. There is no way for the tester to compensate for small errors in tooth spacing between the test and the driving gears. High overload and/or high frequency vibrations result. (2) During operation the gears shift axially owing to creep, wear, etc. The contact surface is continually engaging with unused, unworn surface introducing this unwanted variable. (3) No compensation has been made for tooth deflection. At a unit load of 2000 pounds per inch of face width, the teeth will deflect approximately .001 inch, causing two adverse effects: (a) The tip of the tooth next coming into mesh will interfere with its mating tooth, causing abnormally high tendency to scuff. (b) The longer tooth of the test set will bow causing high stress concentration at the ends of the mating tooth.

Remedying these conditions will increase effectiveness and reliability of the apparatus, and will minimize the error caused by difficult-to-control tooth manufacturing inaccuracy. We recommend the following modifications: (1) Mount test gears on separate shafts from drive gears and connect the two with reasonably flexible quill shafts. (2) Accurately position the test gears with, say, dual ball bearings. (3) Modify addendum of test gears to allow for tooth deflection. (4) Make test gears of equal face width.

As a longer range development, we suggest that further study be given to the nature of scoring. B. W. Kelly (of Caterpillar Tractor Co.) has demonstrated that scoring is most seriously affected by: (1) oil temperature, (2) gear load, (3) rubbing speed, and (4) surface finish. He has demonstrated that: $T_c = T_b + [K_1 W_n (\sqrt{V_p} - \sqrt{V_g})] / \sqrt{b}$, where: T_c — a critical scoring temperature, peculiar to the oil being tested, T_b — gear blank temperature, K_1 — friction factor, a function of gear material and surface finish, W_n — normal unit load at point of scoring, V_g & V_p — rotational velocities of gear and pinion at point of scoring, b — Hertz compressive band width.

The right hand member calculates the temperature rise of the oil film as a particular point on the tooth goes through mesh. If this temperature rise when added to the bulk oil temperature exceeds a critical limit, scoring will occur. Our work has corroborated this fact, although there is some question on the magnitude of K_1.

When the above temperature has been exceeded, scoring will usually proceed rapidly until complete destruction of tooth profile occurs. Consequently we question the choice of 22.5% scuffed surface as a reference point. No doubt this value has been chosen empirically to make test results best match observed field operation.

Consequently, we recommend that the above expression be evaluated on this apparatus. Scuffing tendency may then be measured in terms of temperature, which appears to be basically true, rather than in terms of tooth load which is only one of the variables.

Item	Oil	Gear Pitch	Hertz Stress psi	Unit Load ppi	Oil Temp °F.	PLV	Test Method
1	1010	8	43,000	120	165	11,000	Txco
2	AN03 (MIL-O-6086)	8	193,000	2415	165	11,000	Txco
3	1065	10	140,000	1240	165	9160	Pwa
4	1100	10	175,000	1950	165	9160	Pwa
5	MIL-L-7808	10	172,000	1885	165	9160	Pwa
6	1005	10	225,000	3200	>175	8500	GE
7	1010	10	225,000	3200	>250	8500	GE
8	MIL-L-7808	10	225,000	3200	>310	8500	GE

As shown above, correlation of our experience with various lubricants and data from the gear and lubricant tester has not been good. Lines 1 through 5 were data obtained by the one laboratory on this machine while lines 6, 7 & 8 were obtained from a similar, though more elaborate apparatus in our Gear Engineering Division. The laboratory referred to above, following standard procedure for these tests, lists the load at which 22.5% of the tooth surface was scuffed. Lines 1 & 2 were run with 0.50" face width gears, inspected after each 15 minutes of operation. Lines 3, 4 & 5 employed 0.25" gears, inspected after 10 minute runs. The latter procedure is standard. Lines 6, 7 & 8 are operating conditions that may be consistently maintained on our test stand without scoring or scuffing. The critical dependence of the ratings upon the selection of gear test is obvious.

COMMENTARY by J. M. Stokely, J. G. Carroll & S. R. Calish, California Research Corp.: Mr. Ryder has mentioned the coke test which has evolved as a method of predicting "hot spot" performance of aviation oils. We would like to present some further discussion of this test. The Table below illustrates laboratory devices for evaluating coking. Two main differences are apparent: First, the California Research equipment involves a through-air flow, whereas the Pratt & Whitney unit, which is now incorporated into the MIL-L-7808 specification utilities a reverse air flow. Second, the oil splasher is parallel with the oil level in one case whereas in the specification device the splasher dips into the oil to various depths.

In the coking test, reservoir oil is picked up by the wires on the oil splasher and thrown against the overhead test panel which is maintained at a fixed temperature, usually 600 F. Over a period of time coke deposition occurs, the weight of which is the object of the test. Ideally, this deposition is, of course, held to a minimum.

Fluid	Calif. Research Equipt.	Pratt & Whitney Equipt.	Appearance of Test Engine
AN-0-9 Grade 1010	354	400	Coked
A	96	382	Very clean
B	207	100	Clean
C	217	575	Marginal coking

Table I Comparative data from two testers. Panel coking tests, 1000 rpm, 600 F., 10 hours, deposition in milligrams.

To interpret these results, it is necessary to know what factors influence the coke test. In the case of a non-additive mineral oil, such as the AN-0-9, Grade 1010, the coke deposit represents deterioration of the base oil which results in a fairly high coke deposit, both in the engine and in the bench test. In the case of the three synthetic base lubricants, A, B, & C, the coke test shows only 25 to 35 mg. deposit when the base synthetics are tested. Additives in the finished synthetic fluids, which are included to increase bulk oxidation stability, deactivate metal catalysts, reduce corrosion on bearing materials, and increase the film strength to permit higher gear loadings, are responsible for most of the coke deposit measured on these fluids. Deposits which may be induced by excessive contact with air in the bench test may remain on the bench test panel, but these deposits may never form or may be harmlessly dissolved in the engine.

There is evidence that diester base lubricants, such as Fluid B, give a clean engine when deposits are around the 100 mg. level on the Pratt & Whitney coke test and a dirty engine when

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the deposit approaches the 200 mg. level. On the other hand, a polyglycol base fluid, such as Fluid A, gives a clean engine with a coke test of 382 mg. on the Pratt & Whitney equipment, but a similar polyglycol base material with different additives, Fluid C, gives a coke test of 575 mg. and is marginal on engine coke. It may be necessary to consider both the type of fluid as well as the coke deposited in order to correlate with engine performance, unless changes in the coke test can be devised to reflect engine results more accurately.

By further study and modification it may be possible to develop a coke test which rates different synthetic fluids in the same way as the various engines rate them. Until then, the coke test should not be used as a final basis for rejection of otherwise promising new lubricants. This test performs a useful function in research on new lubricants and in quality control to insure uniformity of qualified lubricants. It does not yet correlate well enough with various engines and various base materials to be a safe basis for rejection of new types of fluids.

A rating system for oils may be desirable, but it would be extremely difficult to reconcile the conflicting requirements of various engines. Some engines may require special resistance to coking; other engines may require high film strength for prop-jet gears. In some cases exceptional low temperature properties may be the quality required above all others in order to permit proper functioning of hydraulically actuated mechanisms using the central oil system. At the present stage of rapid engine design changes and with the potential introduction of many new classes of synthetic lubricant bases, an equitable rating system would be difficult to devise.

(SECTION NEWS, from page 194)
Annual election of officers (See ASLE Directory), together with an address by C. C. Mugford, Southern Pacific Co., entitled Fingerprinting Used Diesel Engine Oil. Mr. Mugford was concerned with predicting the mechanical condition of railroad diesel motive power units by spectrographic examination of the crankcase oil. Previous methods of acquiring standards by wet analysis and preparing each oil sample by weighing, ashing, reweighing and transferring the residue to the electrode, were found to be too lengthy. A new method in which known amounts of soluble lithium and cadmium compounds were added to the oil sample as internal standards required only one weighing. A rotating electrode dipping in this standardized oil sample in a helium atmosphere permitted it to be shot in the spectrograph without further treatment. The method was simple enough to permit examining oil samples from all equipment every thirty days, in addition to the existing viscosity, sludge content and water content checks at the end of each trip. It was found that the detergency level could be accurately determined from the calcium and barium concentrations shown by the spectrograph. Since one brand of crankcase oil was used throughout, concentrations of other elements could be compared from sample to sample as long as the detergency levels were similar. Settling and filtering did not seem to greatly affect the oil analysis as long as the dispersing power of the detergent remained. Once engine construction details were learned and a record of samples from each engine had been accumulated, prediction of most engine failures became possible. High lead concentration was traceable to excessive main bearing wear, silver to a particular type of silver-plated wrist pin, chromium to water leaks which permitted the chromate inhibitor in the coolant to invade the crankcase, and silicon to faulty air cleaners. The record to date shows that some mechanical breakdowns en route still occur, but that in all cases where engine overhaul has been ordered because of spectrographic evidence, the trouble was actually found in th engine.

May meeting — J. T. Clark, Vickers, Inc., presented a paper entitled Hydraulic System Requirements And The Importance Of Proper Lubricants, pointing out that hydraulic systems were being used to clamp, to position, and to supply power. Generally they consisted of a reservoir, filter, cooler, pump and a means of control. Fluids in common use were water, which has the disadvantage of causing corrosion and high wear rates; petroleum oils, which reduce rust and wear but may form gum or sludge under some conditions; and the recently introduced synthetics with a wide range of properties. In general the fluid must not only match the assumptions made during the design of the equipment, but also take cognizance of the conditions under which the equipment is operating . Contaminnants found in hydraulic systems generally were (1) solids, either resulting from system wear and corrosion, carried in with the fluid, or inhaled; (2) other liquids, generally water; and (3) air sucked in at various low pressure spots in the system. Reservoir design was considered the principal factor in preventing and removing contamination. It must provide conditions in which fluid turbulence can subside and permit separation of solids, other fluids, and air. Fuild properties considered were viscosity and viscosity index, pour point, neutralization number, corrosion inhibition, lubricity, compressibility, stability, and emulsibility. (Sub-

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mitted by R. S. McCord, Sec'y-Treasurer.)

MILWAUKEE: May meeting — Annual installation of officers (See ASLE Directory. Submitted by R. W. Schroeder, Sec'y.)

NEW YORK: May meeting — Annual election of officers (See ASLE Directory), together with an address by W. Schaefer, The Lubrizol Corp., entitled Additives — Their Use & Evaluation In Lubricating Oils, in which the various types of additives presently used were explained, and evaluation test procedures for internal combustion engine applications were covered. (Submitted by E. Landau, Treasurer.)

NORTHERN CALIFORNIA: May meeting — W. G. Beggs, Stewart-Warner Corp., presented a report on the methods of application of oil mist lubrication to the multitude of different types of industrial machinery. The objectives of continuous and uniform lubrication were discussed, and this automatic and centralized lubrication technique, with reference to its effect on production, was evaluated.

June meeting — Informal dinner dance at the Naval Officer's Club, Alameda Naval Air Station. (Submitted by P. M. Ruedrich, Sec'y-Treasurer.)

ONTARIO: May meeting — Annual election of officers (See ASLE Directory. Submitted by W. B. Dembner, Retiring Sec'y.)

PHILADELPHIA: May meeting— Annual election of officers (See ASLE Directory), together with a symposium on dry film lubrication. E. E. Bisson, NACA Lewis Flight Prop. Lab., discussed the coefficiency of friction from the standpoint of research and development; M. L. Bunting, Acheson Colloids Co., discussed collodial gravity and its many uses from the standpoint of the manufacturer or supplier; and E. E. Weismantel, Westinghouse Electric Corp., discussed the various types of dry film lubrication from the user's standpoint. (Submitted by J. L. Beatty.)

PITTSBURGH: May meeting — Annual election of officers (See ASLE Directory. Submitted by A. A. Raimondi, Retiring Chairman.)

TWIN CITIES: May meeting —

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Membership in the Society is in several grades as defined below. Assignment to grade is made by the Admissions Committee or Board of Directors on the basis of information submitted or supplied by references. MEMBERS: Members shall be: (1) persons not less than twenty-four years in age who are engaged in research and instruction at technical schools, universities, and various publicly and privately supported institutions in the field of lubrication; or (2) persons not less than twenty-four years in age who have occupied recognized positions as lubrication engineers for a period of three or more (not necessarily consecutive) years prior to date of admission, involving the responsibility for or supervision of the development, selection, field use and application of lubricants as differentiated from other activities; or (3) persons not less than twenty-four years in age who are indirectly concerned with the field of lubrication, but possessing other qualifications of experience, knowledge, and accomplishment, have manifested a particular interest in the purposes and welfare of the Society, to the extent that their membership would be a valuable contribution to the successful functioning of its activities. Fee $12.50.

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INDUSTRIAL MEMBERS: Industrial Members are such persons or organizations as may be interested in and desire to contribute to supporting the purposes and activities of the Society. Fee $150.00.

Annual election of officers (See ASLE Directory), together with an address by C. D. Johnson, Twin-City Testing & Engineering Lab., entitled Lube Oil Additives — a practical explanation of the many additives used in today's industrial and automotive lube oils. The meeting concluded with a technicolor sound film entitled Behind The Checkered Flag, courtesy Socony-Vacuum Oil Co., Inc., covering the Indianapolis 500 mile auto race. (Submited by B. T. Harding, Chairman, and W. H. Furst, Retiring Sec'y.)

WHEELING: May meeting—Annual election of officers (See mile auto race. (Submitted by W. P. Liller, Retiring Sec'y-Treasurer.)

YOUNGSTOWN: April meeting — Annual election of officers (See ASLE Directory. Submitted by J. A. Samuels, Sec'y.)

(LUBE IN THE NEWS, from page 200)

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THE RIGHT QUANTITY

Knowledge of the lubricant capacity of a bearing need not rely upon mathematical or theoretical calculations. Injector Valves are externally adjustable to accurately deliver *the exact quantity* of lubricant required by each bearing.

THE RIGHT TIME

The pumping unit forces lubricant to all bearings in the system, simultaneously, at the specified time interval by means of the selective setting of the pressure developed by the central pump.

ADEQUATE LUBRICANT FILM

A bearing lubricant film can be maintained constantly by the control possible in timing the frequency of lubricant replenishment.

CONTAMINATION

Contamination of lubricant is eliminated when dispensing from an original refinery container or self-contained reservoir through the sealed injector circuit, direct to the bearing.

Apply the RIGHT LUBRICANT.
In the RIGHT QUANTITY.
At the RIGHT TIME.

LINCOLN ENGINEERING COMPANY

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LUBRICANT APPLICATION SYSTEMS

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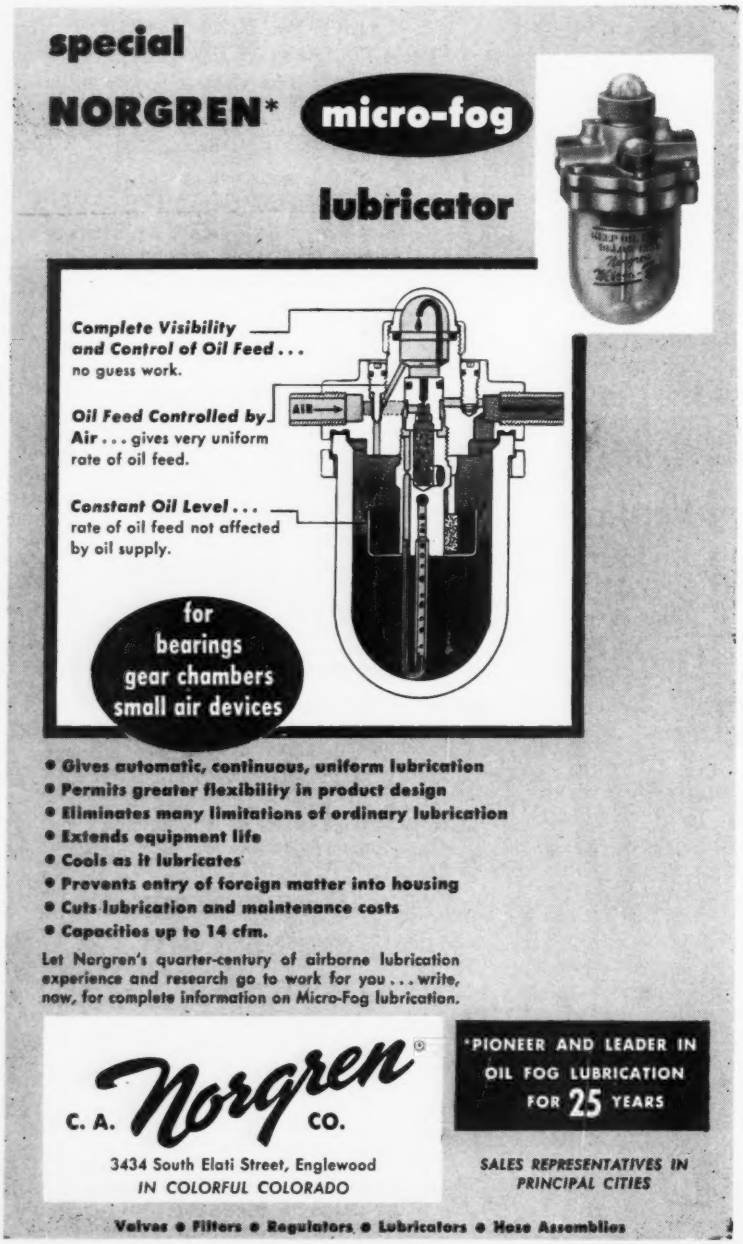
Resisting pressures of more than 250,000 lbs. psi, and temperature extremes from minus 90 F., to plus 750 F., it is not affected by solvents, ordinary chemicals or corrosive liquids. By filling surface irregularities, it reduces friction and wear, protects pressure points, enables more power to be used productively instead of for overcoming resistance, materially adds to machine and tool life, and reduces maintenance costs. It is supplied in a variety of carriers, volatile, non-volatile, and solid in diverse molds. For further information, write Hill Dale Industries, Inc., 57 E. 73rd St., New York 21, N. Y.

"J" TYPE COOLANT CLARIFIER, a recently developed compact, multiple filter-tube unit, features over 16,000 square inches of filtering area; removes all chips, abrasives, dirt and other solid contaminants from both water based and mineral oil coolants used in individual machine tools or small central systems. Clarification of coolant is accomplished by passing liquid through multiple filtering tubes suspended vertically within the clarifier. Self-cleaning tubes, washed free of contaminants during backwash period, require no manual cleaning or maintenance; the unit can be operated continuously, independent of the machine tool, or intermittently, as desired. Several models are available depending on capacities to be handled and individual filtering requirements. For complete information, write Honan-Crane Corp., 818 Wabash Ave., Lebanon, Ind.

LAMINATED DISC FILTERS: An expendable, laminated fiber disc recharge with four to ten times the useful life of cellulose, waste or redwood, has been recently designed for the full flow filtering of large quantities of oil. Classed as an extended area recharge because its actual filtering area is greater than the area of its container, in a recent test on a 1000 HP supercharged diesel engine at an oil company's pipe line station, the recharges kept oil cleaner (0.02% by volume) for a period of 3500 engine hours or nearly 12 times as long as the cotton waste recharges ordinarily used. The laminated discs remove foreign solids smaller than two microns and the absorption of these solids after 48 hours of draining at 70 F. is 2½ times the weight of the recharge; because no chemicals, bleaches or diatomaceous earths are used, the units will not remove vital additives from detergent oils. For complete details, write W. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.

METALLOID WOS BULLETIN, recently released, describes applications and methods for the use of the new Dynatomics water-and-oil soluble metal cutting fluid said to function on a heat-limiting principle; suitable for all metal removal operations, including grinding, honing, cut threading, broaching, milling, shaping, turning, drilling, boring, and tapping, the bulletin also describes Metalloid's modified tool geometry program involving the use of 18-degree relief angles on single point tools in conjunction with the WOS metal cutting fluid. For a copy of this bulletin, write Metalloid Corp., Huntington, Ind.

MULTISTAGE CENTRIFUGAL BLOWERS & EXHAUSTERS, for 1 to 9 psi pressures or vacuum from 2 to 12 ins. mercury, are described in a new bulletin listing seventeen industrial applications,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Let Norgren's quarter-century of airborne lubrication experience and research go to work for you . . . write, now, for complete information on Micro-Fog lubrication.
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Valves • Filters • Regulators • Lubricators • Hose Assemblies

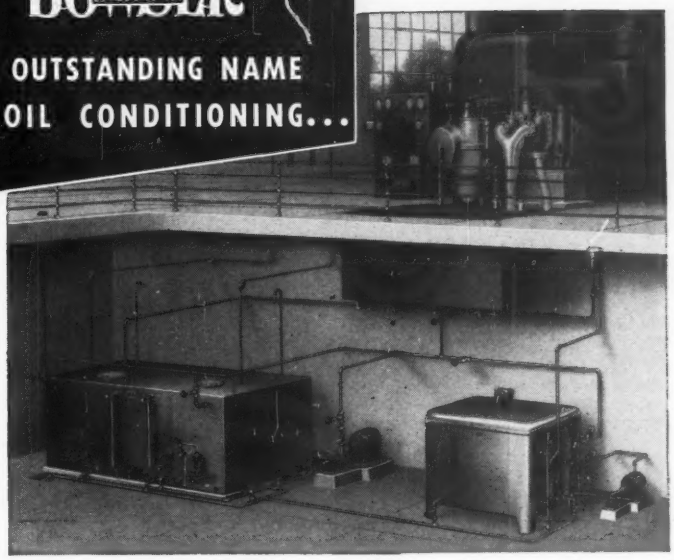
including several on which savings can be made by replacing high-cost compressed air formerly considered necessary. Also included are typical performance curves and check lists of data required for optimum blower or exhauster specification. For copies of this bulletin, write U. S. Hoffman Machinery Corp., 105 Fourth Ave., New York 3, N. Y.

PILOT-CONTROLLED AIR PRESSURE REGULATOR, recently announced, has been designed for extreme precision in air control over a wide operating range — from 2 psi to 120 psi with a capacity to deliver more than 200 cfm. Providing greater flexibility and efficiency in machine design and plant layout, and a more convenient setup for plant personnel, the pressure regulator can be placed where it will be most effective and the separate pilot-control regulator can be installed at the most convenient location. Utilizing regulated air pressure to control delivered air pressure, the pilot-controlled regulator minimizes the effect of fluctuating line pressure as a factor in delivering uniform working air pressure. Available in ½", ¾" and 1" sizes, a light adjusting spring in the pilot-control regulator makes regulation easy and precise and eliminates the heavy adjusting springs of other type regulators. For additional data, write C. A. Norgren Co., 3434 S. Elati St., Englewood, Colo.

ROCKWELL BULLETIN V-220: A new, comprehensive 12-page bulletin describing lubricants, lubricant fittings, and lubrication methods for Rockwell-built Nordstrom valves has been issued by the Rockwell Mfg. Co. Three major types of lubricant — Rockwell Hypermatic, Nordcoseal, and Lubricant DC-234 — are described, and their functions listed in detail. Illustrated and briefly described are the new bulk and gun tube lubricants along with stick lubricants, bulk lubricant cans and Hypreseal stem packing for use in the packing injector fittings of Nordstrom Hypreseal valves. A special feature outlines the three functions of a lubricant in Nordstrom valves — lubrication of the seat, provision of an impervious seal around each port under pressure, and provision of the hydraulic means for lifting the plug from its tapered seat.

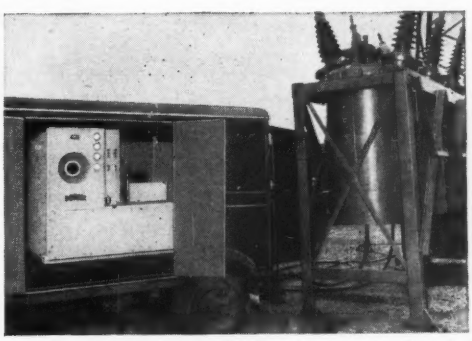
BOWSER ESTABLISHED 1885

THE OUTSTANDING NAME IN OIL CONDITIONING...



...for power generating equipment!

This Bowser turbine oil conditioning system keeps oil "bone dry" and crystal-clear . . . the best "stay-on-the-job" insurance a turbine bearing can have. Costly bearing failures are avoided and less stand-by equipment is required with Bowser oil conditioners. Turbine oil lasts for years, too! That's why Bowser oil conditioners are on the job in a majority of power plants.



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The Bowser Hydro-Volifier is a conditioning unit for transformer, circuit breaker and switch oils. It restores oil dielectric to 35 k.v. in a single pass while the equipment remains energized . . . no service interruptions with Bowser oil conditioning!

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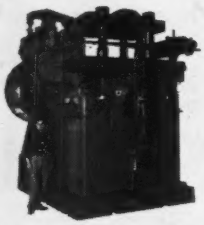

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—say HENRY & WRIGHT of Hartford, Conn.

"While waiting for delivery of one of our 25-ton Dieing Machines to do a particularly heavy job, a customer was trying to start production of a 10-ton machine. Even though it was lubricated with a conventional grease every eight minutes, the machine had to be shut down for bearings to cool during each coil run. Then, on our recommendation, he changed to a LUBRIPLATE Lubricant. With but two applications of LUBRIPLATE a day, the machine operated continuously except during change of coils."

For nearest LUBRIPLATE distributor, see Classified Telephone Directory. Send for free 56-page "LUBRIPLATE DATA BOOK"... a valuable treatise on lubrication. Write LUBRIPLATE DIVISION, Fiske Brothers Refining Co., Newark 5, N. J. or Toledo 5, Ohio.

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For complete details, write Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa., Bulletin V-220.

SOLNUS OILS, newly designed to give more lubrication per dollar, have excellent time-temperature characteristics and can be applied by any method. Physical preperties and advantages are dicussed in a technical bullentin which may be obtained by writing Sun Oil Co., Philadelphia 3, Pa., Bulletin 17.

Patent Abstracts

Prepared by ANN BURCHICK from OFFICIAL GAZETTE, Vol. 669, No. 4; Vol. 670, Nos. 1, 2, 3, 4; Vol. 671, Nos. 1, 2, 3, 4. Printed copies of patents are available from the Patent Office at twenty-five cents each. Address the Commissioner of Patents, Washington, D. C. for copies and for general information concerning patents.

MINERAL OIL ADDITIVE, Patent #2,636,858.
by R. H. Jones & L. E. Moody, assignors to Standard Oil Development Co.
A mineral oil containing about 0.1 to about 20% by weight of an additive consisting of (A) a product obtained by reacting a phosphorus sulfide with a hydrocarbon and neutralizing the acidic reaction product thus formed with a nitrogen base, and (B) a zinc dithiocarbamate.

COMPOSITION OF MATTER, Patent #2,636,859
by G. Entwistle & H. J. Voss, assignors to Sinclair Refining Co.
A grease composition consisting essentially of a mineral oil and soap compounded grease and about 0.05 to about 2 weight percent of an oil soluble higher alkylated diphenylamine and about 0.05 to 2 weight percent of a dimethylaminomethyl alkylphenol.

LUBRICANT & HYDRAULIC FLUID COMPOSITIONS, Patent #2,636,862
by F. J. Watson, assignor to Shell Development Co.
A hydraulic fluid composition comprising a major proportion of a dialkylarylphosphine oxide, 0.25 to 5% by weight of the total composition of an epoxy compound and 0.025 to 1% by weight of the total composition of a sulfur-containing organic compound selected from the group consisting of aliphatic disulfides, aromatic disulfides, sulfurized sperm oils, and sulfurized terpenes.

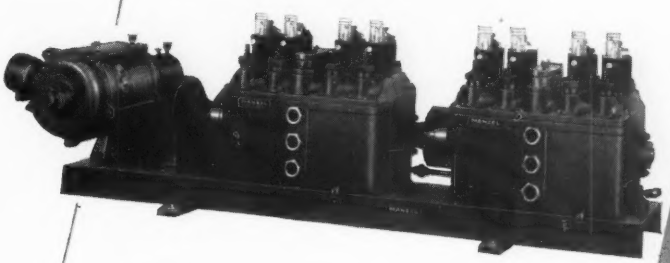
OIL-BASE DRILLING FLUIDS, Patent #2,637,692.
by P. G. Nahin, assignor to Union Oil Co. of California.
A drilling fluid composition essentially comprising a major proportion of a mineral oil and a minor proportion, suf-

HOW TO CURE

"Fluid-Feedosis"*

A CASE HISTORY

PROBLEM: Chemical Companies needed really dependable equipment for high pressure lubrication of compressor cylinders and bearings.



SOLUTION: Manzel developed a completely new line of heavy duty Force Feed Lubricators that pump S. A. E. 40 oil at pressures up to 30,000 PSIG. Enthusiastic reports of performance are being received from users.

*Difficulties in pumping chemicals, oils, and other fluids in accurate quantities for lubrication, chemical treating, or other purposes.

For the solution to lubrication difficulties, or chemical feeding problems, more and more companies are turning to Manzel. For 50 years a leader in the field, Manzel is today a flexible, fast-moving organization with the special technical skill for meeting *your* needs quickly and economically. Write for further information about Manzel Force Feed Lubricators and Chemical Feeders.

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Safeguard high priced machinery with MANZEL FORCE FEED LUBRICATION

- Reduce Wear
- Prevent Breakdown
- Slash oil consumption
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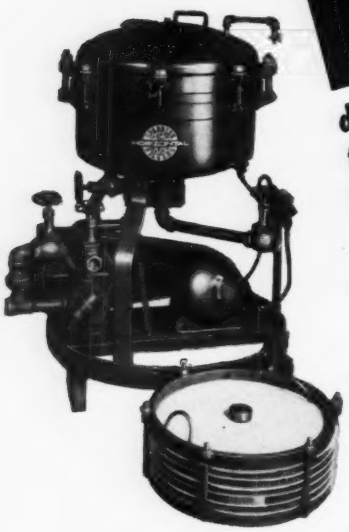
Keep your hydraulic oils, and coolants clean with . . .

SPARKLER FILTERS

designed specifically for this purpose. A practical protection for hydraulic machinery and cutting tools



The VR filter is designed for filtering hydraulic oil, quenching oils, transformer oil, solvents and other industrial liquids.

★ Clean hydraulic oil greatly prolongs the life of all moving parts, such as pistons, cylinders, pumps, valves, etc.

★ Filtering hydraulic oil extends the life of the oil. Drain and leakage oil may be reclaimed. One large user of hydraulic machinery saved $90,000.00 in hydraulic oil alone in one year by filtering oil with Sparkler Filters.

★ Filtering hydraulic oil eliminates sticking and clogging of valves and reduces down time and maintenance cost. A prominent authority on hydraulic machinery says that 70% of servicing and repairs is due to the improper condition of hydraulic oil.

★ Filtering coolant fluid removes all fine metal particles and grit from grinding wheels that is pumped back to nozzles unless a filter is employed in cleaning up the return coolant.

★ Clean filtered coolant prevents scratching and flat spots on work piece and increases the life of grinding wheels and cutting tools. Reduces the frequency of dressing grinding wheels.

The Sparkler VR filter is constructed on a simple filtering principle using filter paper as a media. The filter plates can be removed easily for cleaning which consists only of removing the dirty paper and replacing with fresh sheets. Any shopman can make this change of paper in a few minutes. The cost of replaced paper is less than a dollar per change.

The Sumptype filter is ideal for filtering coolants. The filter plate assembly is attached to the intake pipe of the circulating pump and the coolant is drawn through the filter discs and delivered clean and free from all metal particles and grit, to the nozzels.

Let a Sparkler representative demonstrate these filters in your plant

SPARKLER MANUFACTURING CO.
Mundelein, Ill.

REPRESENTATIVES IN ALL PRINCIPAL CITIES

ficient to impart gel strength and wall-building properties to said oil of a modified clay prepared by heating a hydratable clay at a temperature between about 250 and about 500 C. for from about 0.5 to about 10 hours, cooling the clay and forming a slurry of the cooled clay and an agent selected from the class consisting of glycols and alkyl ethers of glycols containing from 2 to 10 carbon atoms, allowing said slurry to stand until said agent is adsorbed into the inter-lamina spaces of the clay, and thereafter drying the clay containing the adsorbed agent.

YARN LUBRICANT, Patent #2,637,693
by J. L. Baggett & M. W. Duke, assignors to Celanese Corp.

A composition for the treatment of textile materials having a basis of cellulose acetate, which comprises a two-component composition consisting of (1) a mixture of 50 to 60 parts by weight of mineral oil, 4 to 12 parts by weight of tricresyl phosphate, 5 to 14 parts by weight of oleic acid, 2.5 to 7 parts by weight of triethanolamine and 0.25 to 0.75 part by weight of oleyl alcohol, and (2) 20 to 30 parts by weight of a conditioning agent comprising the desalted and dehydrated sodium hydroxide neutralized reaction product of oleum with a mixture of oleic acid, peanut oil and mineral oil, having diamyl phenol incorporated therein, which reaction product has been diluted with about 60% by weight of mineral oil, the two-component composition having a water content of about 1.3 to 1.6% by weight in equilibrium with the water content of the atmosphere at a humidity of 55 to 65%.

GREASE COMPOSITIONS, Patent #2,637,694.
by W. H. Peterson & T. Skei, assignors to Shell Development Co.

A grease composition comprising a major proportion of a lubricating oil, a minor amount at least sufficient to impart a grease structure to said composition of a soap of a higher hydroxy fatty acid and from about 1% to about 20%, based on the weight of said soap, of magnesium hydroxide.

CALCIUM-SODIUM SOAP GREASES FROM HIGHLY OXIDIZED WAXES, Patent #2,637,695
by J. K. McKinley, B. A. Scott, W. H. Goff & R. F. Nelson, assignors to the Texas Co.

A lubricating grease comprising essentially a mineral oil fraction of lubricating characteristics as the predominating constituent, and about 10-40% by weight of a mixed calcium-sodium soap of a saponifiable wax oxidate obtained by oxidizing a petroleum wax under controlled conditions to produce an oxidation product having a neutralization number in the range of from about 100 to about 300 and a saponification number in the range of from about 175 to about 450, said grease having a dropping point of at least 482 F., the proportion of sodium soap in said mixed calcium-sodium soap being at least 40% by weight.

AROMATIC-FREE MINERAL OIL RUST INHIBITING COMPOSITIONS, Patent #2,637,703.
by T. W. Dixon & L. W. Sproule, assignors to Standard Oil Development Co.

A corrosion inhibiting composition comprising a substantially aromatic-free mineral oil containing dissolved therein an oil-soluble sodium petroleum sulfonate in an amount of 15 to 30 weight per cent, ethyleneglycol monobutylether in an amount of 0.75 to 1.5 weight per cent, and an agent selected from the group consisting of (1) a mixture of sorbitan monolaurate in an amount of from 3 to 6 weight per cent and sorbitan monooleate. in an amount of 0.75 to 1.5 weight per cent, and (2) pentaerythritol monooleate in an amount of 2 to 4 weight per cent.

Current Literature

ADDITIVES FOR LUBRICATING GREASES
by R. T. Macdonald & J. L. Dreher, Institute Spokesman, Vol. 17, No. 1, Apr. 1953, pp. 16-19, 21, 23-26.
In general, the additives used in greases are of the same type as those used in oils; however, they are usually employed at higher concentrations in greases. The three principal types of additives for greases are rust preventives, extreme pressure agents, and oxidation inhibitors. These are discussed in detail with respect to need, methods of evaluation, and types of effective compounds or materials. The other kinds of additives, such as fillers, coloring agents, additives to eliminate the separation of oil, and stringiness agents, are discussed only in a general way.

APPARATUS FOR MEASUREMENT OF ELECTRICAL PROPERTIES OF COLLOIDAL SUSPENSIONS IN OILS
by C. J. Penther & A. Bondi, Journal of Physical Chemistry, Vol. 57, No. 5, May 1953, pp. 540-541.
A capacitor cell with rotating inner electrode and sensitive conductimeter are used for measurement of flocculation and deflocculation in non-aqueous systems. The speed of the electrode is controlled steplessly and held constant by a thyratron circuit. The motor load current is proportional to torque and is therefore a measure of viscosity. Conventional oscillator, bridge and null detector are used for dielectric constant and loss measurements over the range 20 c./sec. to 100 kc./sec.

ARE YOU LUBRICATING MOTORS PROPERLY?
by F. H. Walker, Jr., & W. J. O'Meara, Petroleum Refiner, Vol. 32, No. 5, May 1953, pp. 127-131.
As result of extensive tests it appears that best lubrication results from lubricating a motor only once every three years, using sealed bearing housings and high quality grease. Motor speed is highly critical. An extension of lubrication intervals makes feasible the elimination of the familiar grease cups and fittings, thereby insuring the motors against either improper lubrication or over-lubrication which, if permitted, can cause considerable harm.

DETERMINATION OF COPPER IN MINERAL OILS USING AN ION-EXCHANGE TECHNIQUE, THE
by H. Buchwald & L. G. Wood, Analytical Chemistry, Vol. 25, No. 4, April 1953, pp. 664-665.
In an investigation requiring the determination of the copper content of a large number of samples of used transformer oil it was found that the generally accepted methods were lengthy, and were not capable of application to low concentrations of copper. It was found that if a cation-exchange resin containing sulfonate groups was converted to the hydrogen form and then washed with 2-propanol to remove water, it would remove dissolved copper compounds from mineral oil quantitatively. The copper could be recovered quantitatively by percolation of dilute sulfuric acid down the column. The copper content of the acid percolate could then be determined colorimetrically.

Coming Events

AUGUST
17-19 Socy. of Automotive Engineers (International West Coast Meeting), Georgia Hotel, Vancouver, B. C., Canada.

SEPTEMBER
6-11 Amer. Chemical Socy. (124th National Meeting), Conrad Hilton Hotel, Chicago, Ill.
9-11 Oil Industry Information Committee, Cleveland Hotel, Cleveland, Ohio.
13-16 Amer. Inst. of Chemical Engineers, Fairmont & Mark Hopkins Hotels, San Francisco, Calif.
14-17 Socy. of Automotive Engineers (National Tractor Meeting & Production Forum), Schroeder Hotel, Milwaukee, Wisc.
16 Amer. Petroleum Inst. (Division of Marketing, Lubrication Committee Meeting), The Traymore Hotel, Atlantic City, N. J.
16-18 National Petroleum Assoc. (51st Annual Meeting), The Traymore, Atlantic City, N. J.
21-25 Instrument Socy. of America (8th National Instrument Conference & Exhibit), Sherman & Morrison Hotels, Chicago, Ill.
23-25 National Industrial Conference Board, Waldorf-Astoria Hotel, New York, N. Y.
27-30 Amer. Socy. of Mechanical Engineers (Annual Conference, Petroleum Division), Rice Hotel, Houston, Tex.
27 to Oct. 2 Amer. Socy. for Testing Materials (Committee D-2 on Petroleum Products & Lubricants), Shoreham Hotel, Washington, D. C.
29 to Oct. 3 Socy. of Automotive Engineers (National Aeronautic Meeting), Statler Hotel, Los Angeles, Calif.

OCTOBER
7-9 National Assn. of Corrosion Engineers, South Central Region (Annual Meeting), Mayo Hotel, Tulsa, Okla.

APRIL 1954
5-7 American Society of Lubrication Engineers (9th Annual Meeting & Exhibit), Netherlands Plaza Hotel, Cincinnati, Ohio.

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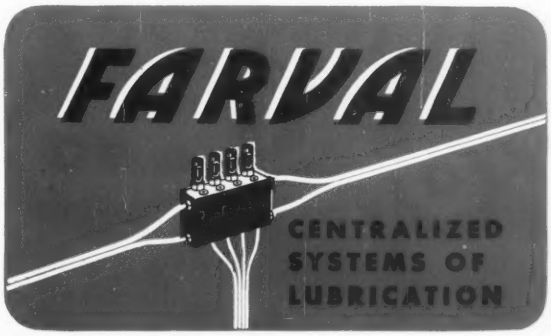

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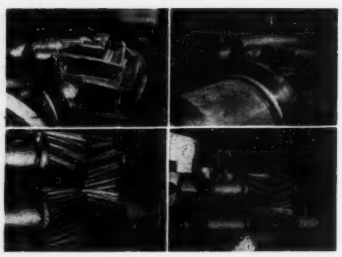

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